THE STEAM ENGINE: A TREATISE ON STEAM ENGINES AND BOILERS

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NEW-YORK.

THE STEAM ENGINE:

A TREATISE
ON STEAM ENGINES AND BOILERS.

COMPRISING THE PRINCIPLES AND PRACTICE OF THE COMBUSTION OF FUEL,
THE ECONOMICAL GENERATION OF STEAM, THE
CONSTRUCTION OF STEAM BOILERS;
AND
THE PRINCIPLES, CONSTRUCTION, AND PERFORMANCE OF STEAM ENGINES—
STATIONARY, PORTABLE, LOCOMOTIVE, AND MARINE,
EXEMPLIFIED IN ENGINES AND BOILERS OF RECENT DATE.

ILLUSTRATED BY ABOVE 1300 FIGURES IN THE TEXT, AND A SERIES OF
FOLDING PLATES, DRAWN TO SCALE.

BY

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The ratio of compression is $\left(\frac{21\% + 7\%}{7\%} = \right)$ 4; and the final pressure of steam in the clearance-space is $17\frac{1}{2}\% \times 4 = 70\%$. The initial work for calculation of compression-resistance is $7\% \times 70\% = 490$; and conducting the calculation in the manner already exemplified,—

	Units.
Work of compression, 490 × (hyp log 4 = 1.386) = ...	679
Deduct, work identified with back-pressure, in terms of work in clearance ($17\frac{1}{2}\% \times 3 \times 7\%$) = ...	367½
Net work of compression............................	311½ (*b*)

The quantity of exhaust-steam reclaimed is the difference of the total steam in the clearance, less the steam of back-pressure already there; or $7\% \times (70\% - 17\frac{1}{2}\% = 52\frac{1}{2}\% =)$ 367½ (*a*).

Ratio of steam *a*, to work *b*, : : 367½ : 311½ : : 1 : .805.

The two ratios of steam to work are here compared:—

A : B : : 1 :	1.390
a : *b* : : 1 :	.805
Difference...............	.585,

the significance of which is to the effect that the net work of compression was less in proportion to the steam reclaimed than the positive net work of the steam, in the ratio of 1.390 to .805; and that the difference, .585, represents the proportional work economized in compression, compared with the work which the reclaimed steam is to give out during the next steam-stroke. This difference represents the work of the reclaimed steam, measured by 52½ per cent of the clearance-space, multiplied by $\frac{.585}{1.390}$; or $52\frac{1}{2}\% \times \frac{.585}{1.390} = 22.1$ per cent; that is to say, the work of 22.1 per cent of initial clearance-steam is thus provided for; and the portions of clearance-steam provided for are as follows:—

	per cent.
Back exhaust-pressure..	17½
Reclaimed..	52½
Equivalent of work economized	22.1
Total provided for......................................	92.1

Showing that, say, 92 per cent of the clearance-steam is provided for, and that (100 − 92 =) 8 per cent is to be credited to the boiler.

2d Case. The same example, except that the steam is cut off at 50 per cent of the stroke, for which the proper period of compression is 17 per cent of the stroke. The initial quantity of steam is (100 × (50 + 7) =) 5700, of which there remains in the clearance $(7\% \times 17\frac{1}{2}\% =)$ 122½. The net steam is equal to (5700 − 122½ =) 5577½ (A). For the initial work and the actual expansion-ratio 1.88, the total work is 9247; and, deducting for passive work in the clearances and back exhaust-pressure, the remainder 6847 (B) is the net work. The ratio of compression is $\left(\frac{17+7}{7}=\right)$ 3.43; (17½%×3.43=) 60 per cent of steam compressed into the clearance. The initial work is (7%×60%=) 420, and the net work of compression is 518 less ((17½%×2.43×7%=) 298) = 220 (*b*). The quantity of steam reclaimed is 298 (*a*).

The two ratios of steam to work are:—

A : B : : 5577½ : 6847 : : 1 : 1.23
a : *b* : : 298 : 220 : : 1 : .74

Difference........... .49

And the portions of clearance-steam provided for are as follows:—

	per cent.
Back exhaust-pressure	17.5
Reclaimed (17½ % × 2.43 =)	42.5
Equivalent to work economized (42.5 × $\frac{.49}{1.23}$ =)	17.0
Total provided for	77.0
To be credited to the boiler	23.0
	100.0

3d Case. A condensing engine, cutting off at 20 per cent of the stroke; with a back-pressure 5 per cent; clearance 7 per cent; proper period of compression 47 per cent. The actual ratio of expansion is 4, and the ratio of compression is $\left(\frac{47+7}{7}=\right)$ 7.71. The two ratios of steam to work are:—

A : B : : 2665 : 5242 : : 1 : 1.97
a : *b* : : 235 : 316 : : 1 : 1.34

Difference......... 0.63

The portions of clearance-steam provided for are as follows:—

	per cent.
Back exhaust-pressure	5
Reclaimed (5 % × 6.71 =)	33.5
Equivalent to work economized (33.5 × $\frac{.63}{1.97}$ =)	10.7
Total provided for	49.2
Credited to boiler	50.8
	100.0

4th Case. The same engine, cutting off at 20 per cent of the stroke; with a back-pressure of 2½ per cent; clearance 7 per cent; and proper period of compression 52 per cent. The actual ratio of expansion is 4, and the ratio of compression is 8.43. The two ratios of steam to work, taken in the same manner as before, are:—

A : B : : 2682 : 5492 : : 1 : 2.05
a : *b* : : 130 : 184 : : 1 : 1.40
Difference........ .65

The portions of clearance-steam provided for are as follows:—

	per cent.
Back exhaust-pressure	2.5
Reclaimed (2½ % × 7.43 =)	18.6
Equivalent for work economized $(18.6 \times \frac{.65}{2.05} =)$	6.0
Total provided for	27.1
To be credited to the boiler	72.9
	100.0

5*th Case*. A compound-cylinder condensing engine, of which in the 1st cylinder the steam is cut off at 50 per cent of the stroke, with a back-pressure 30 per cent; clearance 7 per cent; and proper period of compression 12 per cent. The actual ratio of expansion is 1.88, and the ratio of compression is 2.714. The two ratios of steam to work are:—

A : B : : 5490 : 5577 : : 1 : 1.02
a : *b* : : 360 : 208 : : 1 : .58
Difference......... .44

The clearance-steam is thus provided for:—

	per cent.
Back exhaust-pressure	30
Reclaimed (30 × 1.74 =)	51.4
Equivalent for work economized $\left(51.4 \times \frac{.44}{1.02} =\right)$	22.2
	103.6

Here it appears that the whole of the clearance-steam is provided for and something over (3.6 per cent). For the 2d cylinder, take a period of admission 40 per cent of the stroke; back-pressure 20 per cent; and clearance 7 per cent. The proper period of compression is 18 per cent. The actual ratio of expansion is 2.28; and the ratio of compression is 3.57. The two ratios of steam to work are:—

A : B : : 1 : 1.688
a : *b* : : 1 : .767
Difference .... .921

The clearance-steam is thus provided for:—

	per cent.
Back exhaust-pressure	20
Reclaimed (20 × 2.57 =)	51.4
Equivalent for work economized $(51.4 \times \frac{.921}{1.688} =)$	28
	99.4

Showing that all the clearance-steam is provided for, except an insignificant fraction.

These five cases represent average ordinary types of practice; and they show that, in ordinary non-condensing engines, having 7 per cent clearance, suitable provision may be made, in round numbers, for the equivalent of from 80 to 90 per cent of the initial steam for the clearance, crediting only 10 or 20 per cent of it to the boiler; and that, in single-cylinder condensing engines, from 30 to 50 per cent is provided for, crediting the remainder to the boiler. In two-cylinder condensing compound engines, again, it is shown that the whole of the clearance-steam may be provided for, crediting nothing to the boiler. The effect of such provision is to reduce the quantity of non-effective clearance-steam drawn direct from the boiler—non-effective, that is to say, for initial work—so that the 7 per cent clearance thus provided is virtually equivalent to a clearance of about $\left(\frac{7 \times (100-85)}{100} =\right)$ 1 per cent for non-condensing engines; and about $\left(\frac{7 \times (100-40)}{100} =\right)$ 4 per cent for condensing engines. For compound engines the clearance-space may be wholly neutralized. The reduction of efficiency of the steam by clearance, unneutralized, of which examples have been given, is proportionally lessened.

The means of economizing the expenditure of initial steam for clearance, are applicable, of course, to engines having clearances other than 7 per cent.

This beneficial exploitation of clearance-space is based on the element of back-pressure, for without back-pressure—that is to say, supposing that there were a perfect vacuum—there could not be any compression. But, given the back-pressure, it is better to recognize its existence, and to reclaim a portion of the exhaust-steam, by economically inclosing and compressing a portion of it—minimizing, at the same time, the loss of force incurred by the clearance-space. In extreme cases, it may be imagined, an absolute gain of efficiency might be effected by the agency of clearance, compared with the condition of non-clearance; but such cases could only arise in the presence of a very high exhaust-pressure. Here, one is confronted by the paradoxical conclusion that clearance-space, so far from being necessarily a cause of loss of efficiency, may actually be made accessory to an augmentation of the efficiency of steam in the cylinder; that is to say, it may be made the means of adding to what may be called the normal efficiency under the given conditions, with non-clearance. It may at least be reasonably inferred that it is scarcely worth while to specialize the design of an engine, for the sake of reducing clearance to a minimum. True, the employment of compression causes a resistance which leads to a reduction of the available work done on the piston. In any case, an inch or two added to the diameter of the cylinder, or two or three pounds more pressure of steam, may supply the needful compensation.

CHAPTER II.—RULES AND TABLES FOR WORK OF STEAM IN A SINGLE CYLINDER.

I. TABLE OF RATIOS OF EXPANSION OF STEAM, WITH RELATIVE PERIODS OF ADMISSION, PRESSURES, AND TOTAL PERFORMANCE.

To facilitate calculations about steam expanded in cylinders, the table No. 94 has been composed. The actual ratios of expansion, column 1, range from 1.0 to 8.0, for which the hyperbolic logarithms are given, for ready reference, in column 2. The 3d column contains the periods of admission relative to the actual ratios of expansion, as percentages of the stroke, calculated by Rule 3, page 386. The 4th column gives the values of the mean pressures relative to the initial pressures, the latter being taken as 1, calculated by formula (6), page 384. The 5th column gives the values of the initial pressures relative to the mean pressures, when the latter are taken as 1. These values are the reciprocals of those of the 4th column; at the same time they may be calculated by formula (4), page 384. In the calculation of these last three columns, 3, 4, and 5, clearance is taken into account, and its amount is assumed at 7 per cent of the stroke. In the 6th column, of final pressures, they are such as would be arrived at by the continued expansion of the whole of the steam to the end of the stroke, the initial pressure being equal to 1. They are the reciprocals of the ratios of expansion, column 1, as indicated by Rule 4, page 386.

The 7th column contains the relative total performances of equal weights of steam worked with the several actual ratios of expansion: the total performance when steam is admitted for the whole of the stroke, without expansion, being equal to 1. They are calculated on the principle exemplified at page 387.

The pressures have been calculated on the supposition that the pressure of steam, during its admission into the cylinder, is uniform up to the point of cutting off, and that the expansion is continued regularly to the end of the stroke. The relative performances have been calculated without any allowance for the effect of compressive action.

The calculations have been made for periods of admission ranging from 100 per cent, or the whole of the stroke, to 6.4 per cent, or $^1/_{16}$th of the stroke. And though, nominally, the expansion is 16 times in the last instance, it is actually only 8 times, as given in the first column. The great difference between the nominal and the actual ratios of expansion is caused by the clearance, which is equal to 7 per cent of the stroke, and causes the nominal volume of steam admitted, namely, 6.4 per cent, to be augmented to $6.4+7=13.4$ per cent of the stroke, or, say, double, for expansion. When the steam is cut off at $^1/_9$th, the actual expansion is only 6 times; when cut off at $^1/_5$th, the expansion is 4 times; when cut off at $\frac{1}{3}$d, the expansion is $2\frac{2}{3}$ times; and to effect an actual expansion to twice the initial volume, the steam is cut off at 46½ per cent of the stroke, not at half stroke.

Table No. 94.—EXPANSIVE WORKING OF STEAM:—ACTUAL RATIOS OF EXPANSION; WITH THE RELATIVE PERIODS OF ADMISSION, PRESSURES, AND PERFORMANCE.

Clearance at each end of the cylinder, 7 per cent of the stroke.

(SINGLE CYLINDER.)

1 ACTUAL RATIO OF EXPANSION: Or Number of Volumes to which the Initial Volume is Expanded.	2 HYPERBOLIC LOGARITHM of Actual Ratio of Expansion.	3 CORRESPONDING PERIOD OF ADMISSION, or CUT-OFF. Clearance, 7 per cent. of the Stroke.	4 AVERAGE TOTAL PRESSURE.	5 TOTAL INITIAL PRESSURE.	6 TOTAL FINAL PRESSURE.	7 RATIO OF TOTAL PERFORMANCE of Equal Weights of Steam. (Col. 4÷Col. 6.)
initial volume =1.		stroke = 100.	initial pressure=1.	mean pressure=1.	initial pressure=1.	with 100 % of admission =1.000.
1.0	.0000	100	1.000	1.000	1.000	1.000
1.05	.0488	95.0	.9997	1.003	.952	1.050
1.1	.0953	90.3 or 9/10	.996	1.004	.909	1.096
1.15	.1398	86.0	.990	1.010	.870	1.138
1.18	.1698	83.3 or 5/6	.986	1.014	.847	1.164
1.2	.1823	82.1	.983	1.017	.833	1.180
1.23	.2070	80.0 or 4/5	.980	1.020	.813	1.206
1.25	.2231	78.6	.977	1.024	.800	1.221
1.3	.2624	75.3 or ¾	.969	1.032	.769	1.261
1.35	.3000	72.3	.961	1.041	.741	1.297
1.39	.3293	70.0 or 7/10	.953	1.049	.719	1.325
1.4	.3365	69.4	.951	1.052	.714	1.332
1.45	.3716	66.8 or 2/3	.942	1.062	.690	1.365
1.5	.4055	64.3	.932	1.073	.666	1.399
1.54	.4317	62.5 or ⅝	.925	1.081	.649	1.425
1.55	.4382	62.0	.922	1.085	.645	1.429
1.6	.4700	59.9 or 3/5	.913	1.095	.625	1.461
1.65	.5008	57.9	.903	1.107	.606	1.490
1.7	.5306	56.0	.894	1.119	.588	1.520
1.75	.5595	54.1	.883	1.132	.571	1.546
1.8	.5878	52.4	.873	1.145	.555	1.573
1.85	.6153	50.8	.864	1.157	.541	1.597
1.88	.6314	50.0 or ½	.860	1.163	.532	1.616
1.9	.6419	49.3	.854	1.171	.526	1.624
1.95	.6678	47.9	.846	1.182	.513	1.649
2.0	.6931	46.5	.836	1.196	.500	1.672
2.1	.7419	44.0	.818	1.222	.476	1.718
2.2	.7885	41.6	.799	1.251	.455	1.756
2.28	.8241	40.0 or 2/5	.787	1.271	.439	1.793
2.3	.8329	39.5	.782	1.279	.435	1.798
2.4	.8755	37.6 or ⅜	.766	1.305	.417	1.837
2.5	.9163	35.8	.750	1.333	.400	1.875
2.6	.9555	34.2	.736	1.359	.385	1.912
2.65	.9745	33.3 or 1/3	.726	1.377	.377	1.925
2.7	.9933	32.6	.719	1.391	.370	1.943
2.8	1.030	31.2	.706	1.416	.357	1.978
2.9	1.065	29.9 or 3/10	.692	1.445	.345	2.006
3.0	1.099	28.7	.679	1.473	.333	2.039
3.1	1.131	27.5	.665	1.504	.323	2.059
3.2	1.163	26.4	.652	1.534	.313	2.083
3.3	1.194	25.4	.641	1.560	.303	2.115

Table No. 94 (*continued*).

1 ACTUAL RATIO OF EXPANSION: Or Number of Volumes to which the Initial Volume is expanded.	2 HYPERBOLIC LOGARITHM of Actual Ratio of Expansion.	3 CORRESPONDING PERIOD OF ADMISSION, or CUT-OFF. Clearance, 7 per cent. of the stroke.	4 AVERAGE TOTAL PRESSURE.	5 TOTAL INITIAL PRESSURE.	6 TOTAL FINAL PRESSURE.	7 RATIO OF TOTAL PERFORMANCE of equal Weights of Steam. (Col. 4÷Col. 6.)
initial volume = 1.		stroke = 100.	initial pressure = 1.	mean pressure = 1.	initial pressure = 1.	with 100 % of admission = 1.000.
3.35	1.209	25.0 or ¼	.637	1.570	.298	2.129
3.4	1.224	24.5	.631	1.585	.294	2.146
3.5	1.253	23.6	.619	1.615	.286	2.164
3.6	1.281	22.7	.608	1.645	.278	2.187
3.7	1.308	21.9	.597	1.675	.270	2.211
3.8	1.335	21.2	.589	1.698	.263	2.240
3.9	1.361	20.4	.579	1.727	.256	2.262
4.0	1.386	19.7 or $^1/_5$	.567	1.764	.250	2.278
4.1	1.411	19.1	.559	1.789	.244	2.291
4.2	1.435	18.5	.551	1.815	.238	2.315
4.3	1.459	17.9	.542	1.845	.233	2.326
4.4	1.482	17.3	.533	1.876	.227	2.348
4.5	1.504	16.8 or $^1/_6$	.526	1.901	.222	2.370
4.6	1.526	16.3	.518	1.930	.217	2.387
4.7	1.548	15.8	.511	1.957	.213	2.399
4.8	1.569	15.3	.503	1.988	.208	2.418
4.9	1.589	14.8	.494	2.024	.204	2.422
5.0	1.609	14.4 or $^1/_7$	.488	2.049	.200	2.440
5.2	1.649	13.6	.476	2.101	.193	2.466
5.4	1.686	12.8	.462	2.164	.185	2.497
5.5	1.705	12.5 or $^1/_8$	.457	2.188	.182	2.511
5.6	1.723	12.1	.450	2.222	.178	2.528
5.8	1.758	11.4	.438	2.283	.172	2.547
5.9	1.775	11.1 or $^1/_9$	.432	2.315	.169	2.556
6.0	1.792	10.8	.427	2.342	.167	2.567
6.2	1.825	10.3	.419	2.387	.161	2.585
6.3	1.841	10.0 or $^1/_{10}$	.413	2.421	.159	2.597
6.4	1.856	9.7	.407	2.457	.156	2.609
6.6	1.887	9.2 or $^1/_{11}$	.398	2.513	.152	2.619
6.8	1.917	8.7	.388	2.577	.147	2.639
7.0	1.946	8.3 or $^1/_{12}$	.381	2.625	.143	2.664
7.2	1.974	7.9	.373	2.681	.139	2.683
7.3	1.988	7.7 or $^1/_{13}$	.369	2.710	.137	2.693
7.4	2.001	7.5	.365	2.740	.135	2.703
7.6	2.028	7.1 or $^1/_{14}$	.357	2.801	.132	2.711
7.8	2.054	6.7 or $^1/_{15}$	.348	2.874	.128	2.719
8.0	2.079	6.4 or $^1/_{16}$	.342	2.924	.125	2.736

Though a uniform clearance of 7 per cent at each end of the stroke has been assumed as an average proportion for the purpose of compiling the table, the clearance of cylinders with ordinary slides varies considerably—say, from 5 to 8 or 9 per cent. With the clearance, 7 per cent, that has been assumed, the table gives approximate results sufficient for most practical purposes. It supplies means of economizing calculation, certainly more trustworthy than such results as can be deduced by calculations based on simple tables of hyperbolic logarithms, where clearance is neglected.

It has already been exemplified at page 386, how the table may serve in making approximate calculations when the clearance is other than 7 per cent.

II. TOTAL WORK DONE BY ONE POUND OF STEAM EXPANDED IN A SINGLE CYLINDER.

If 1 pound of water be converted into steam of atmospheric pressure—14.7 lbs. per square inch, or 2116.8 lbs. per square foot—it gradually occupies a volume equal to 26.36 cubic feet; and the work done in acquiring this volume under one atmosphere is equal to 2116.8 lbs. × 26.36 feet = 55,788 foot-pounds. The equivalent quantity of heat expended is 1 unit per 772 foot-pounds, or, altogether, (55,788 ÷ 772 =) 72.3 units. This is precisely the work of 1 pound of steam of one atmosphere, acting on a piston without expansion.

The gross work thus done on a piston by 1 pound of steam, generated at total pressures varying from 15 lbs. to 100 lbs. per square inch, varies, in round numbers, from 56,000 to 62,000 foot-pounds, equivalent to from 72 to 80 units of heat.

The simple work of a pound of steam, without expansion, thus exemplified, is reduced by clearance according to the proportion it bears to the net capacity of the cylinder. If the clearance be 7 per cent of the stroke, then 107 parts of steam are consumed in doing the work of a stroke, which is represented by 100 parts, and the work of a given weight of steam without expansion, admitted for the whole of the stroke, is reduced in the ratio of 107 to 100. Having determined by this ratio the quantity of work by 1 pound of steam without expansion, as reduced by clearance, the work of the same weight of steam for various ratios of expansion may be deduced from that, in terms of the relative performance of equal weights of steam, as exemplified at page 389, and given in the 7th column of table No. 94. The simple work, without expansion, as reduced by clearance, is to be multiplied by the ratio or relative performance just referred to. The simple work of a pound of steam does not greatly vary with the pressure; and, for present purposes, the work of steam of a total pressure of 100 lbs. per square inch has been calculated and tabulated. This pressure corresponds to a net pressure, above the atmosphere, of 85 lbs. per square inch—a convenient average standard of pressure. The volume of 1 pound of saturated steam of 100 lbs. per square inch is 4.33 cubic feet, and the pressure per square foot is 144 × 100 = 14,400 lbs.; then, the total simple work—or total initial work, as it may be called—is,

$$14{,}400 \times 4.33 = 62{,}352 \text{ foot-pounds.}$$

This amount is to be reduced for a clearance of, say, 7 per cent, thus:—

$$62{,}352 \times \frac{100}{107} = 58{,}273 \text{ foot-pounds,}$$

which is the total simple work of 1 pound of steam of 100 lbs. total pressure per square inch, after the loss by clearance is deducted; and, divided by

Joule's equivalent, 772, it is equal to 75.5 units of heat. Now, the total or constituent heat of 1 pound of 100-lb. steam, reckoned from a temperature of 212° F., is 1001.4 units; reckoned from 102° F., the temperature of water from the condenser under a pressure of 1 lb. per square inch, the constituent heat is 1111.4 units. The equivalent of the net simple work, 75.5 units, is, then, 7.5 per cent of the total heat reckoned from 212° F., or 6.7 per cent if reckoned from 102° F. For shorter admissions, with complementary expansion, the work is increased as in the following examples:—

When the steam is cut off at,

1, ¾, ½, ¼, 1/5, ⅛, 1/10, 1/15 of stroke,

the actual ratios of expansion are,

1, 1.3, 1.88, 3.35, 4.0, 5.5, 6.3, 7.8 times;

the comparative performances of 1 pound of steam are as

1, 1.261, 1.616, 2.129, 2.278, 2.511, 2.597, 2.719,

and the total actual work of 1 lb. of 100-lb. steam is, in the same proportion,

58,273, 73,513, 94,200, 124,066, 132,770, 146,325, 151,370, 158,414 foot-pounds.

The equivalents, as heat, of the actual work done, are

75.5, 95.2, 122.0, 160.7, 171.9, 189.5, 196.1, 205.2 units,

which are, in parts of the constituent heat reckoned from 102° F., equal to

6.7, 8.5, 11.0, 14.5, 15.5, 17.0, 17.6, 18.5 per cent.

From these examples, it appears that the total work done by 1 pound of steam, without making any allowance for back-pressure, compression, or other contingencies, varies from about 60,000 foot-pounds when applied without expansion, to about double that, or about 120,000 foot-pounds, when expanded three times, cutting off at about 27 per cent of the stroke; and to about 150,000 foot-pounds, or 2½ times the first performance, when expanded about six times, cutting off at about 10 per cent of the stroke.

Also that, of the heat consumed in the formation of steam, not 7 per cent is converted into total work when there is no expansive action; that with an expansion of six times there is only 17½ per cent converted; and that even with an expansion of eight times, when the steam is cut off at 1/15th, less than 20 per cent, or one-fifth of the heat consumed, is converted into work. The remainder of the heat is lost, as for the purpose of the steam engine.

III. CONSUMPTION OF STEAM WORKED EXPANSIVELY PER HORSE-POWER OF TOTAL WORK PER HOUR.

The measure of a horse-power is the performance of 33,000 foot-pounds per minute, or of 33,000 × 60 = 1,980,000 foot-pounds per hour. This work is to be divided by the work of 1 lb. of steam, and the quotient is the weight of steam or water required per horse-power per hour. For example, the total actual work done in the cylinder by 1 lb. of 100-lb. steam, without expansion and with 7 per cent of clearance, is 58,273 foot-pounds; and $\frac{1{,}980{,}000}{58{,}273} = 34$ lbs. of steam, is the weight of steam consumed for the total work done in the cylinder per horse-power per hour. For any shorter period of admission, with expansion, the weight of steam per horse-power is less, as the total work by 1 lb. of steam is more, and may be found by dividing 1,980,000 foot-pounds by the respective total work done; or by dividing 34 lbs. by the ratio of performance, column 7, table No. 94. In this way it is found that, when the steam is cut off at

1, ¾, ½, ¼, 1/5, ⅛, 1/10, 1/15 of stroke,

the quantities of steam, or water as steam, consumed per horse-power of total work per hour, are

34.0, 26.9, 21.0, 16.0, 14.9, 13.5, 13.1, 12.5 lbs.

Further, allowing that 10 lbs. of steam are generated by the combustion of 1 lb. of coal, the fuel consumed per horse-power of total work per hour is,

3.40, 2.69, 2.10, 1.60, 1.49, 1.35, 1.31, 1.25 lbs.

IV. TABLE OF THE TOTAL WORK DONE BY 1 POUND OF STEAM OF 100 LBS. TOTAL PRESSURE PER SQUARE INCH.

The table, No. 95, which follows, is calculated on the basis of the conditions above laid down, which are repeated under the heading of the table, for ready reference. The 1st, 2d, and 3d columns are repeated from table No. 94. The 4th column, of total actual work done by 1 lb. of steam of 100 lbs. total pressure, is calculated by multiplying the work without expansion, namely, 58,273 foot-pounds, by the ratios in column 3, for the proportional work when expanded. The 5th column contains the equivalent of heat converted into work, which is found by dividing the work in foot-pounds by Joule's equivalent, 772; and the 6th and 7th columns give these values as percentages of the total heat of steam raised from 212° and 102° F. respectively. The 8th column contains the quantity of steam consumed for the total work done per horse-power per hour.

An initial total pressure of 100 lbs. per square inch has been adopted for the table, No. 95, as an average pressure in ordinary good practice, and the contents of the table are good as approximate values for other pressures considerably different from 100 lbs., more or less. A tablet, page 413, is appended to the table, containing multipliers for various other total pressures, which may be applied to the total actual work given in the table for the purpose of determining the correct total quantities of work for steam of the respective pressures. These multipliers are arrived at by multiplying the total pressure of any other given steam per square foot, by the volume in cubic feet of 1 lb. of such steam, and dividing the product by 62,352, which is the product in foot-pounds for steam of 100 lbs. pressure. The quotient is the multiplier for the given pressure. From the tablet it appears that, between the extremes of 65 lbs. and 160 lbs. per square inch, the deviation from the work done as given for 100 lbs. pressure does not exceed 2½ and 3 per cent.

Table No. 95.—TOTAL WORK DONE BY ONE POUND OF STEAM OF 100 LBS. TOTAL PRESSURE PER SQUARE INCH.

ASSUMPTIONS.—That the initial pressure is uniform; that the expansion is complete to the end of the stroke; that, substantially, the pressure in expansion varies inversely as the volume; that there is no back-pressure of exhaust or of compression.

Volume of 1 lb. of steam of 100 lbs. pressure per square inch, or 14,400 lbs. per square foot........4.33 cubic feet.
Product of initial pressure and volume........62,352 foot-pounds.
Constituent heat of 1 lb. of this steam—Reckoned from 212° F........1001.4 units.
Reckoned from 102° F........1111.4 units.
Clearance at each end of the cylinder........7 per cent of the stroke.

ACTUAL RATIO OF EXPANSION.	CORRESPONDING PERIOD OF ADMISSION or CUT-OFF, in percentage of Stroke.	TOTAL ACTUAL WORK DONE by 1 lb. of 100-lb. Steam.		EQUIVALENT OF HEAT converted into Work.			Quantity of Steam consumed per Horse-power of actual Work done per Hour.
		Ratio of Work done (col. 7, table No. 94).	Actual Work done.	Heat converted.	Percentage of Constituent Heat converted, as calculated from 212° F. and 102° F.		
(1) initial vol. = 1.	(2) per cent.	(3)	(4) foot-pounds.	(5) units.	(6) % from 212° F.	(7) % from 102° F.	(8) lbs.
1.0	100	1.000	58,273	75.5	7.5	6.7	34.0
1.05	95	1.050	61,193	79.3	7.9	7.1	32.4
1.1	90.3 or 9/10	1.096	63,850	82.7	8.3	7.5	31.0
1.15	86.0	1.138	66,310	85.9	8.6	7.8	29.9
1.18	83.3 or 5/6	1.164	67,836	87.9	8.8	7.9	29.2
1.2	82.1	1.180	68,766	89.1	8.9	8.0	28.8
1.23	80.0 or 4/5	1.206	70,246	91.0	9.1	8.2	28.2
1.25	78.6	1.221	71,151	92.2	9.2	8.3	27.8
1.3	75.3 or ¾	1.261	73,513	95.2	9.5	8.5	26.9
1.35	72.3	1.297	75,575	97.9	9.8	8.8	26.2
1.39	70.0 or 7/10	1.325	77,242	100.1	10.0	9.0	25.6
1.4	69.4	1.332	77,616	100.6	10.1	9.1	25.5
1.45	66.8 or 2/3	1.365	79,555	102.9	10.3	9.3	24.9
1.5	64.3	1.399	81,546	105.6	10.6	9.5	24.3
1.54	62.5 or ⅝	1.425	83,055	107.6	10.8	9.7	23.8
1.55	62.0	1.429	83,299	107.9	10.8	9.7	23.7
1.6	59.9 or 3/5	1.461	85,125	110.3	11.0	9.9	23.3
1.65	57.9	1.490	86,828	112.5	11.3	10.2	22.8
1.7	56.0	1.520	88,598	114.8	11.5	10.4	22.4
1.75	54.1	1.546	90,115	116.7	11.7	10.5	22.0
1.8	52.4	1.573	91,680	118.7	11.9	10.7	21.6
1.85	50.8	1.597	93,065	120.5	12.0	10.8	21.3
1.88	50.0 or ½	1.616	94,200	122.0	12.2	11.0	21.0
1.9	49.3	1.624	94,610	122.5	12.3	11.1	20.9
1.95	47.9	1.649	96,100	124.5	12.4	11.2	20.6
2.0	46.5	1.672	97,432	126.2	12.6	11.3	20.3
2.1	44.0	1.718	100,266	129.7	13.0	11.7	19.8
2.2	41.6	1.756	102,366	132.5	13.2	11.9	19.4
2.28	40.0 or 2/5	1.793	104,466	135.3	13.5	12.2	19.0
2.3	39.5	1.798	104,855	135.7	13.6	12.3	18.9
2.4	37.6 or ⅜	1.837	107,050	138.6	13.9	12.5	18.5
2.5	35.8	1.875	109,266	141.5	14.1	12.7	18.1
2.6	34.2	1.912	111,400	144.3	14.4	13.0	17.8

Table No. 95 (*continued*).

ACTUAL RATIO OF EXPANSION.	CORRESPONDING PERIOD OF ADMISSION or CUT-OFF, in percentage of Stroke.	TOTAL ACTUAL WORK DONE by 1 lb. of 100-lb. Steam.		EQUIVALENT OF HEAT converted into Work.			Quantity of Steam consumed per Horse-power of actual Work done per Hour.
		Ratio of Work done (col. 7, table No. 94).	Actual Work done.	Heat converted.	Percentage of Constituent Heat converted, as calculated from 212° F. and 102° F.		
(1) initial vol. = 1.	(2) per cent.	(3)	(4) foot-pounds.	(5) units.	(6) % from 212° F.	(7) % from 102° F.	(8) lbs.
2.65	33.3 or 1/3	1.925	112,220	145.4	14.5	13.1	17.7
2.7	32.6	1.943	113,244	146.7	14.7	13.2	17.6
2.8	30.2	1.978	115,244	149.2	14.9	13.4	17.2
2.9	29.9 or 3/10	2.006	116,885	151.4	15.1	13.6	16.9
3.0	28.7	2.039	118,820	153.9	15.4	13.9	16.7
3.1	27.5	2.059	119,970	155.4	15.5	13.9	16.5
3.2	26.4	2.083	121,386	157.2	15.7	14.1	16.3
3.3	25.4	2.115	123,278	159.6	16.0	14.4	16.1
3.35	25.0 or ¼	2.129	124,066	160.7	16.1	14.5	16.0
3.4	24.5	2.146	125,066	162.0	16.2	14.6	15.8
3.5	23.6	2.164	126,125	163.4	16.3	14.7	15.7
3.6	22.7	2.187	127,450	165.1	16.5	14.9	15.5
3.7	21.9	2.211	128,860	166.9	16.7	15.0	15.4
3.8	21.2	2.240	130,533	169.1	16.9	15.2	15.2
3.9	20.4	2.262	131,800	170.7	17.1	15.4	15.0
4.0	19.7 or 1/5	2.278	132,770	171.9	17.2	15.5	14.9
4.1	19.1	2.291	133,500	172.9	17.3	15.6	14.8
4.2	18.5	2.315	134,900	174.8	17.5	15.8	14.7
4.3	17.9	2.326	135,555	175.6	17.6	15.8	14.6
4.4	17.3	2.348	136,825	177.2	17.7	15.9	14.5
4.5	16.8 or 1/6	2.370	138,130	178.8	17.9	16.1	14.34
4.6	16.3	2.387	139,100	180.2	18.0	16.2	14.23
4.7	15.8	2.399	139,800	181.1	18.1	16.3	14.16
4.8	15.3	2.418	140,920	182.5	18.2	16.4	14.05
4.9	14.8	2.422	141,210	182.8	18.3	16.5	14.03
5.0	14.4 or 1/7	2.440	142,180	184.2	18.4	16.6	13.92
5.2	13.6	2.466	143,720	186.2	18.6	16.9	13.78
5.4	12.8	2.497	145,525	188.5	18.8	16.9	13.60
5.5	12.5 or ⅛	2.511	146,325	189.5	18.9	17.0	13.53
5.6	12.1	2.528	147,320	190.8	19.1	17.2	13.44
5.8	11.4	2.547	148,390	192.2	19.2	17.3	13.34
5.9	11.1 or 1/9	2.556	148,940	192.9	19.3	17.4	13.29
6.0	10.8	2.567	149,586	193.7	19.4	17.5	13.23
6.2	10.3	2.585	150,630	195.1	19.5	17.6	13.14
6.3	10.0 or 1/10	2.597	151,370	196.1	19.6	17.6	13.08
6.4	9.7	2.609	152,033	196.9	19.7	17.7	13.02
6.6	9.2 or 1/11	2.619	152,595	197.7	19.8	17.8	12.98
6.8	8.7	2.629	153,810	199.2	19.9	17.9	12.87
7.0	8.3 or 1/12	2.664	155,200	201.1	20.1	18.1	12.75
7.2	7.9	2.683	156,330	202.6	20.3	18.3	12.66
7.3	7.7 or 1/13	2.693	156,960	203.3	20.3	18.3	12.61
7.4	7.5	2.703	157,560	204.1	20.4	18.4	12.57
7.6	7.1 or 1/14	2.711	157,975	204.6	20.4	18.4	12.53
7.8	6.7 or 1/15	2.719	158,414	205.2	20.5	18.5	12.50
8.0	6.4 or 1/16	2.736	159,433	206.5	20.7	18.6	11.83

APPENDIX TO TABLE No. 95.

TABLET OF MULTIPLIERS for the total actual Work done by 1 pound of Steam of other Pressures than 100 lbs. per Square Inch, to be applied to the total actual Work as given in the table. See explanatory notice of this tablet, page 410.

Total Pressures below 100 lbs. per Square Inch.		Total Pressures above 100 lbs. per Square Inch.	
lbs. per square inch.	multiplier.	lbs. per square inch.	multiplier.
65	.975	100	1.000
70	.981	110	1.009
75	.986	120	1.011
80	.988	130	1.015
85	.991	140	1.022
90	.995	150	1.025
95	.998	160	1.031

V. NET CYLINDER-CAPACITY RELATIVE TO THE STEAM EXPENDED AND WORK DONE IN ONE STROKE.

The quantity of cylinder-capacity required for the total performance of a given weight of steam admitted for one stroke depends on the volume of the steam, and on the ratio of expansion. If the given weight admitted be multiplied by the volume of 1 pound of the steam and by the actual ratio of expansion, the product is the gross cylinder-capacity, including clearance. For example, if 1 pound of steam of 100 lbs. pressure per square inch be admitted for the whole stroke, without expansion, the gross capacity is the volume of 1 pound of such steam, namely, 4.33 cubic feet; and the net capacity, supposing the clearance to be 7 per cent of the stroke, is

$$4.33 \times \frac{100}{100+7} = 4.047 \text{ cubic feet.}$$

If, again, 2 pounds of steam of 100 lbs. pressure be admitted and expanded into three times its initial volume, the gross capacity is $4.33 \times 2 \times 3 = 25.98$ cubic feet; and the net capacity is

$$25.98 \times \frac{100}{107} = 24.28 \text{ cubic feet.}$$

From this is derived the following rule for net capacity:—

RULE 5.—*To find the net Capacity of Cylinder for a given Weight of Steam admitted for one stroke, and a given actual ratio of expansion.*—Multiply the volume of 1 pound of the steam by the given weight in pounds, and by the actual ratio of expansion. Multiply the product by 100, and divide by 100 plus the percentage of clearance. The quotient is the net capacity of cylinder.

Again, the quantity of cylinder-capacity required for the performance of a given amount of total actual work, in one stroke, depends on the initial pressure, and the actual ratio of expansion, according to the following rule:—

RULE 6.—*To find the net Capacity of Cylinder for the performance of a given amount of total Actual Work, in one stroke, with a given initial pressure,* *and actual ratio of expansion.*—Divide the given work by the total actual work done by 1 pound of steam of the same pressure, and with the same actual ratio of expansion; the quotient is the weight of steam necessary to do the given work, for which the net capacity is found by Rule 5, preceding.

Note 1. Conversely, the weight of steam admitted per cubic foot of net capacity, for one stroke, is the reciprocal of the cylinder-capacity per pound of steam, as obtained by Rule 5.

2. Likewise, the total actual work done per cubic foot of net capacity, for one stroke, is the reciprocal of the cylinder-capacity per foot-pound of work done, as obtained by Rule 6.

3. The total actual work done per square inch of piston per foot of the stroke, is $^1/_{144}$th part of the work done per cubic foot; a prism 1 inch square and 1 foot long being $^1/_{144}$th part of a cubic foot. The work, in either measure, is in direct proportion to the average total pressure per square inch.

4. The resistances of back-pressure of exhaust and of compression are to be added to the net work required to be done, to find the total actual work.

VI. TABLE OF RELATIONS OF NET CAPACITY OF CYLINDER TO STEAM ADMITTED AND TOTAL ACTUAL WORK DONE.

The table No. 96 gives the net capacity of cylinder required in relation to the quantity of steam of 100 lbs. total pressure per square inch consumed, and of total actual work done, in one stroke. Columns 1, 2, and 3, are the ratios of expansion, periods of admission, and total actual work done by 1 lb. of steam of 100 lbs. pressure, repeated from columns 1, 2, and 4, in the previous table, No. 95. In the 4th column are the net capacities of cylinder required for each pound of steam admitted for one stroke, found by Rule 5; and in the 5th column are the net capacities of cylinder for 100,000 foot-pounds of total actual work done in one stroke, found by Rule 6. The 6th column contains the weights of steam of 100 lbs. pressure admitted to the cylinder for one stroke, per cubic foot of net capacity. These values are simply the reciprocals of those in column 4. The 7th column contains the total actual work done by steam of 100 lbs. initial pressure, for one stroke, per cubic foot of net capacity. These values are the products of the reciprocals of those in column 5, by 100,000, since this is the number of foot-pounds for which the values in column 5 are calculated. The 8th column gives the total actual work per square inch of piston-area, per foot of stroke, in foot-pounds; the initial pressure is 100 lbs., and the following pressures, for the different ratios of expansion, may also be read as percentages of the initial pressure. The total actual works are directly proportional to the average total pressures per square inch, as given in column 4, table No. 94, page 406, where the initial pressure is taken as 1; and the former have been found by multiplying the latter respectively by 100.

Tablet of Multipliers.—The contents of the table No. 96 are calculated for steam of 100 lbs. per square inch, total initial pressure; but they are available, with the aid of a set of multipliers, for other pressures. For column 3, the total actual work done, the multipliers have already been

Table No. 96.—NET CYLINDER-CAPACITY, WITH RELATION TO STEAM ADMITTED, AND TOTAL ACTUAL WORK DONE.

For Steam of 100 lbs. total pressure per square inch.

Clearance at each end of the Cylinder, 7 per cent of the stroke.

ACTUAL RATIO OF EXPANSION.	PERIOD OF ADMISSION, or CUT-OFF, as a Percentage of Stroke.	TOTAL ACTUAL WORK DONE by 1 lb. of 100-lb. Steam.	Net Capacity of Cylinder.		Per Cubic Foot of Net Capacity of Cylinder.		TOTAL ACTUAL WORK DONE per Sq. Inch of Piston, per Foot of Stroke, by 100 lbs. Steam.
			Per pound of 100-lb. Steam, admitted in one Stroke.	Per 100,000 Foot-pounds of Total Actual Work done by Steam of 100 lbs. pressure, in one Stroke.	WEIGHT OF STEAM of 100 lbs., Total Pressure admitted for one Stroke, per Cubic Foot.	TOTAL ACTUAL WORK DONE by Steam of 100 lbs., Total Initial Pressure, in one Stroke, per Cubic Foot.	
(1) initial volume = 1.	(2) per cent.	(3) foot-pounds.	(4) cubic feet.	(5) cu. feet.	(6) pound.	(7) foot-pounds.	(8) foot-pounds.
1.0	100	58,273	4.05	6.94	.247	14,400	100
1.05	95.0	61,193	4.25	6.95	.235	14,388	99.97
1.1	90.3 or 9/10	63,850	4.45	6.97	.225	14,347	99.6
1.15	86.0	66,310	4.65	7.02	.215	14,245	99.0
1.18	83.3 or 5/6	67,836	4.78	7.04	.209	14,204	98.6
1.2	82.1	68,766	4.86	7.06	.206	14,164	98.3
1.23	80.0 or 4/5	70,246	4.98	7.09	.201	14,104	98.0
1.25	78.6	71,151	5.06	7.11	.198	14,065	97.7
1.3	75.3 or ¾	73,513	5.26	7.16	.190	13,966	96.9
1.35	72.3	75,575	5.46	7.23	.183	13,831	96.1
1.39	70.0 or 7/10	77,242	5.63	7.28	.178	13,736	95.3
1.4	69.4	77,616	5.67	7.30	.176	13,699	95.1
1.45	66.8 or 2/3	79,555	5.87	7.38	.170	13,550	94.2
1.5	64.3	81,546	6.07	7.45	.165	13,423	93.2
1.54	62.5 or ⅝	83,055	6.23	7.50	.161	13,333	92.5
1.55	62.0	83,299	6.27	7.53	.159	13,280	92.2
1.6	59.9 or 3/5	85,125	6.47	7.61	.155	13,141	91.3
1.65	57.9	86,828	6.68	7.69	.150	13,004	90.3
1.7	56.0	88,598	6.88	7.77	.145	12,870	89.4
1.75	54.1	90,115	7.08	7.92	.141	12,626	88.3
1.8	52.4	91,680	7.30	7.95	.137	12,579	87.3
1.85	50.8	93,065	7.49	8.04	.134	12,438	86.4
1.88	50.0 or ½	94,200	7.61	8.08	.131	12,376	86.0
1.9	49.3	94,610	7.69	8.13	.130	12,300	85.4
1.95	47.9	96,100	7.89	8.22	.127	12,165	84.6
2.0	46.5	97,432	8.09	8.31	.124	12,034	83.6
2.1	44.0	100,266	8.50	8.49	.118	11,778	81.8
2.2	41.6	102,366	8.90	8.70	.112	11,494	79.9
2.28	40.0 or 2/5	104,466	9.23	8.83	.108	11,325	78.7
2.3	39.5	104,855	9.31	8.89	.107	11,249	78.2
2.4	37.6 or ⅜	107,050	9.71	9.07	.103	11,025	76.6
2.5	35.8	109,266	10.12	9.26	.099	10,799	75.0
2.6	34.2	111,400	10.52	9.45	.095	10,582	73.6
2.65	33.3 or 1/3	112,220	10.72	9.56	.093	10,460	72.6
2.7	32.6	113,244	10.93	9.65	.091	10,363	71.9
2.8	30.2	115,244	11.33	9.83	.088	10,173	70.6
2.9	29.9 or 3/10	116,855	11.74	10.04	.085	9,960	69.2

Table No. 96 (*continued*).

ACTUAL RATIO OF EXPANSION.	PERIOD OF ADMISSION, or CUT-OFF, as a Percentage of Stroke.	TOTAL ACTUAL WORK DONE by 1 lb. of 100-lb. Steam.	Net Capacity of Cylinder.		Per Cubic Foot of Net Capacity of Cylinder.		TOTAL ACTUAL WORK DONE per Sq. Inch of Piston, per Foot of Stroke, by 100 lbs. Steam.
			Per pound of 100-lb. Steam, admitted in one Stroke.	Per 100,000 Foot-pounds of Total Actual Work done by Steam of 100 lbs. pressure, in one Stroke.	WEIGHT OF STEAM of 100 lbs., Total Pressure admitted for one Stroke, per Cubic Foot.	TOTAL ACTUAL WORK DONE by Steam of 100 lbs., Total Initial Pressure, in one Stroke, per Cubic Foot.	
(1) initial volume=1.	(2) per cent.	(3) foot-pounds.	(4) cubic feet.	(5) cu. feet.	(6) pound.	(7) foot-pounds.	(8) foot-pounds.
3.0	28.7	118,820	12.14	10.22	.082	9,785	67.9
3.1	27.5	109,970	12.55	10.46	.080	9,560	66.5
3.2	26.4	121,386	12.95	10.67	.077	9,372	65.2
3.3	25.4	123,278	13.35	10.83	.075	9,234	64.1
3.35	25.0 or ¼	124,066	13.56	10.93	.074	9,149	63.7
3.4	24.5	125,066	13.76	11.00	.073	9,091	63.1
3.5	23.6	126,125	14.16	11.21	.071	8,961	61.9
3.6	22.7	127,450	14.57	11.43	.069	8,749	60.8
3.7	21.9	128,860	14.97	11.60	.067	8,621	59.7
3.8	21.2	130,533	15.38	11.78	.065	8,489	58.9
3.9	20.4	131,800	15.78	11.98	.063	8,347	57.9
4.0	19.7 or 1/5	132,770	16.19	12.19	.062	8,203	56.7
4.1	19.1	133,500	16.59	12.39	.060	8,071	55.9
4.2	18.5	134,900	17.00	12.60	.059	7,936	55.1
4.3	17.9	135,555	17.40	12.80	.058	7,812	54.2
4.4	17.3	136,825	17.81	13.01	.056	7,686	53.3
4.5	16.8 or 1/6	138,130	18.21	13.19	.055	7,581	52.6
4.6	10.3	139,100	18.62	13.38	.054	7,474	51.8
4.7	15.8	139,800	19.02	13.58	.053	7,364	51.1
4.8	15.3	140,920	19.43	13.79	.051	7,252	50.3
4.9	14.8	141,210	19.83	14.01	.050	7,138	49.4
5.0	14.4 or 1/7	142,180	20.23	14.23	.049	7,027	48.8
5.2	13.6	143,720	21.04	14.64	.047	6,831	47.6
5.4	12.8	145,525	21.85	15.02	.046	6,658	46.2
5.5	12.5 or ⅛	146,325	22.25	15.20	.045	6,579	45.7
5.6	12.1	147,320	22.66	15.38	.044	6,502	45.0
5.8	11.4	148,390	23.47	15.80	.043	6,329	43.8
5.9	11.1 or 1/9	148,940	23.87	16.01	.042	6,246	43.2
6.0	10.8	149,586	24.28	16.23	.041	6,161	42.7
6.2	10.3	150,630	25.09	16.63	.040	6,013	41.9
6.3	10.0 or 1/10	151,370	25.49	16.83	.039	5,942	41.3
6.4	9.7	152,033	25.90	17.04	.038	5,868	40.7
6.6	9.2 or 1/11	152,595	26.71	17.47	.037	5,724	39.8
6.8	8.7	153,810	27.52	17.89	.036	5,590	38.8
7.0	8.3 or 1/12	155,200	28.33	18.27	.035	5,473	38.1
7.2	7.9	156,330	29.14	18.64	.0343	5,365	37.3
7.3	7.7 or 1/13	156,960	29.54	18.83	.0339	5,311	36.9
7.4	7.5	157,560	29.95	19.01	.0334	5,260	36.5
7.6	7.1 or 1/14	157,975	30.76	19.47	.0325	5,136	35.7
7.8	6.7 or 1/15	158,414	31.57	19.93	.0317	5,018	34.8
8.0	6.4 or 1/16	159,433	32.38	20.31	.0309	4,923	34.2

given in the tablet appended to the table at page 413, for pressures of from 65 lbs. to 160 lbs. per square inch. For column 4, of the net capacity of cylinder per pound of steam expended in one stroke, the multipliers are simply the ratios of the volume of 1 pound of 100-lb. steam to the respective volumes of 1 pound of steam of other initial pressures. Thus, for steam of 65 lbs. total pressure, of which the volume of 1 pound is 6.49 cubic feet, to be compared with 4.33 cubic feet, which is the volume of a pound of 100-lb. steam, the multiplier is—

$$\frac{6.49}{4.33} = 1.5;$$

and the net capacity of cylinder per pound of 65-lb. steam, when the steam is admitted for the whole of the stroke, is

$$4.05 \text{ cubic feet (for 100-lb. steam)} \times 1.5 = 6.07 \text{ cubic feet.}$$

APPENDIX TO TABLE NO. 96.

TABLET OF MULTIPLIERS FOR NET CYLINDER-CAPACITY, STEAM ADMITTED, AND TOTAL ACTUAL WORK DONE.

For Steam of other pressures than 100 lbs. per square inch.

Total Pressures per Square Inch.	MULTIPLIERS.				
	For Column 3. Total Work.	For Column 4. Capacity.	For Column 5. Capacity.	For Column 6. Weight of Steam.	For Columns 7 and 8. Work.
lbs.					
65	.975	1.50	1.54	.666	.65
70	.981	1.40	1.43	.714	.70
75	.986	1.31	1.33	.763	.75
80	.988	1.24	1.25	.806	.80
85	.991	1.17	1.18	.855	.85
90	.995	1.11	1.11	.901	.90
95	.998	1.05	1.05	.952	.95
100	1.000	1.00	1.00	1.00	1.00
110	1.009	.917	.909	1.09	1.10
120	1.011	.843	.833	1.17	1.20
130	1.015	.781	.769	1.28	1.30
140	1.022	.730	.714	1.37	1.40
150	1.025	.683	.667	1.46	1.50
160	1.031	.644	.625	1.55	1.60

For column 5, of table No. 96, the net capacity of cylinder per 100,000 foot-pounds; of total actual work done in one stroke, the capacity for other pressures is modified, in the first place, in the inverse ratio of the multipliers, as found for the tablet, page 413, for 100-lb. steam, and steam of the given pressure; secondly, in the ratio of the volume of 1 pound of 100-lb. steam, to that of a pound of the given steam. For example, for steam of 65 lbs. total pressure per square inch, the weight of 100-lb. steam to do a given total work, as compared with the weight of 65-lb. steam, is as .975 to 1.000, and the volumes of a pound each of the two steams are respectively 4.33 and 6.49 cubic feet, or as 1 to 1.5 as already found. The values in column 5 are therefore to be increased in the compound ratio of

.975 to 1.000
and 1 to 1.5

or, combined, as .975 to 1.5,
or as 1 to 1.54.

The multiplier for 65-lb. steam is thus 1.54.

For column 6, of the weights of steam per cubic foot of net capacity for one stroke, the weights given for 100-lb. steam are to be multiplied by the reciprocals of the multipliers for column 4, corresponding to other pressures. Thus, for 65-lb. steam, the multiplier is the reciprocal of 1.5, or .666. For column 7, of the total actual work done, per cubic foot of net capacity, for one stroke, the work given for 100-lb. steam is to be multiplied by the reciprocal of the multiplier for column 5, as determined for the given other pressure. Thus, for 65-lb. steam, the multiplier is the reciprocal of 1.54, or .65.

For the total actual work done per square inch of piston per foot of stroke, by steam of any other pressure, the values given in column 8, for 100-lb. steam, are to be multiplied by the given pressure and divided by 100. For 65-lb. steam, for example, the multiplier is, in fact, $\frac{65}{100} = .65$, the same as for column 7. Otherwise regarded, the values in column 8 may be taken as percentages of the work for steam as admitted for the whole of the stroke, whatever the initial pressure may be.

It is apparent that the multipliers for columns 7 and 8, are simply equal to the respective total pressures divided by 100, since the multiplier for 100-lb. pressure is taken as 1. These multipliers must obviously be in the direct ratio of the pressures; and, of course, the multipliers for column 5 must be in the inverse ratio of the pressures.

CHAPTER III.—COMPOUND STEAM ENGINES.

I. INTRODUCTION.

When it is designed to expand the steam used in the steam engine into several times its initial volume, a means of conveniently and effectively doing so, is the development of the expansion in a second cylinder, connected with the first, whence it may be exhausted into the condenser. Obviously, the expansion may be further developed in a third cylinder, or, in short, in any number of cylinders successively. The combination of two cylinders for this purpose is known as a compound steam engine.

The compound steam engine, consisting of two cylinders, is reducible to two forms, in which, first, the steam from the first cylinder is exhausted direct into the second cylinder, as in the Woolf engine; and, second, the steam from the first cylinder is exhausted into an intermediate reservoir, whence the steam is supplied to, and expanded in, the second cylinder, as in the "receiver-engine."

To elucidate simply the leading principles of the compound engine, it is assumed that there is no clearance at either end of either cylinder; that there is no frictional resistance nor other hindrance in the engine to the flow of steam; that the pressure of expanding steam is inversely as the volume; that the expansion of the steam is continued in both cylinders to the end of the stroke; and that in the second cylinder there is a perfect vacuum on the exhaust face of the piston. It is assumed, further, that no intermediate fall, or "drop" of pressure takes place between the first and second cylinders; that is, that the initial pressure in the second cylinder is equal to the final pressure in the first cylinder; also, that there is no loss of steam by condensation within the cylinders.

That the desired expansion from the first into the second cylinder may take place, the second cylinder is necessarily of greater capacity than the first, to afford room for the expansion of the steam. If the capacities of the cylinders were equal to each other, it is clear that the first cylinder-full of steam would only be transferred to another space of the same volume. No useful work would be done by this operation, for though the translated steam would press forwards on the second piston, according to its elasticity, it would likewise press backwards with equal force on the first piston, and so counteract itself. To generate useful work by expansion, therefore, in exhausting steam from the first into the second cylinder, the second cylinder must be of greater capacity than the first.

Taken generally, the work done by expansion into the second cylinder, is that due to the augmentation of volume of the steam in this, the second stage of its action, or to the number of times that the volume of the first cylinder is contained in the volume of the second cylinder;—in other words, to the ratio of expansion into the second cylinder. If there be no expansive working of steam in the first cylinder, so that the whole of the expansion is done in the second cylinder, then the work of the steam is calculated on the ratio of the volume of the second to that of the first cylinder. But, if the steam be cut off in the first cylinder before the end of the stroke, the total ratio of expansion is the product of the ratio of expansion in the first cylinder, into the ratio of the volume of the second to that of the first cylinder; in short, the product of the two ratios of expansion.

For example, let the areas of the first and second cylinders be in the proportion of 1 to 3½, the strokes being equal; the capacities are in the same proportion, and the ratio of expansion from the first into the second cylinder is 3½. Further, let the steam be cut off in the first cylinder at half-stroke, or so as to expand the steam to twice its initial volume when the stroke is completed, then the ratio of expansion in the first cylinder is 2, and the total or combined expansion of the steam in the two cylinders is (3½ × 2 =) 7 times the initial volume. The ratios may be formulated thus:—

Expansion in the 1st cylinder,......................................	1 to 2
Do. do. 2d do.	1 to 3½
Total or combined expansion, the product of the two ratios,...	1 to 7

II. WOOLF STEAM ENGINE, WITHOUT CLEARANCE.

To trace the pressure and the expansive action of steam in compound cylinders, begin with the simplest engine,—the Woolf engine,—in which the cylinders are side by side, and the pistons move simultaneously, in the same direction. The first and second cylinders, A and B, in the example, fig. 166, are in section; they have the same length of stroke, and their areas and capacities are as 1 to 3. The pistons, *a* and *b*, are shown to have arrived at the end of their downward strokes. The steam that occupied the first cylinder, below the piston, has been exhausted from the lower end of that cylinder into the upper part of the second cylinder, above the second piston, as indicated by the arrows. At the same time, fresh steam has been admitted into the first cylinder, during one-third of the stroke, and expanded above the piston to the end of the stroke. These movements of the steam are indicated by the diagrams traced above the pistons, which represent the graduated pressures of the steam in both cylinders during the downward stroke of their pistons. The period of admission in the first cylinder is measured by *cd*, one-third of the stroke *cf*; and the diameter of the cylinder *mc*, is adopted as the measure of the total initial pressure of the steam admitted, whilst the side of the cylinder, *mn*, equal to the length of the stroke, is the base, or vacuum line, of the diagram of pressure. The piston moves on from *d* to *f*, whilst the steam expands behind it, and the diminishing expansive pressure is indicated by the expansion-curve *dg*; the terminal pressure of the steam being measured by the ordinate *ng*. The shaded area thus defined shows the variations of pressure upon the advancing piston thoughout the downward stroke. At the end of the stroke, the steam above the piston, having the final pressure *ng*, is exhausted into the lower end of the second cylinder. By the upward strokes of the pistons, the whole of the steam in the first cylinder is transferred to and is made to occupy the second cylinder, below the second piston, enlarged in volume in proportion as the capacity of the second cylinder is greater than that of the first. The pressure, according to the hyperbolic law, is reduced in the same ratio; and, as against the first piston, it follows the expansion-curve of back-pressure, *gh*, of which—

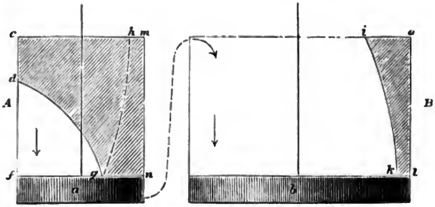


Fig. 166.—Woolf Engine, without Clearance: Ideal Diagram of Action of Steam.

the final exhaust-ordinate *mh*,
is to the initial exhaust-ordinate *ng*, at the commencement of the up-stroke,
as the capacity of the 1st cylinder,
is to the capacity of the 2d cylinder.

The expansion-curve of positive pressure, exerted simultaneously below the second piston, during the up-stroke, is, as before explained, identical with the curve of back-pressure in the first cylinder. For the down-stroke, also, the expansion-curve of positive pressure, represented by *ik*, is an exact counterpart of *gh*, the expansion-curve in the first cylinder, supposing that the distribution and the pressure of the steam are the same for both ends of the cylinder. The curve *ik* shows, in fact, the pressure exerted by the continued expansion of steam into the second cylinder from the lower end of the first cylinder; the initial pressure *oi* in the second cylinder being equal to *ng*, the final pressure in the first cylinder, and the final pressure *lk* in the former being equal to *mh*, the final exhaust-pressure in the latter.

With the data supplied by the diagrams of pressure, the work done by the steam in the two cylinders may be calculated. To find the effective work in the first cylinder, the area of back-pressure, *gm*, is deducted from the area of total pressure, *nc*; the remainder, *ghcd*, is the effective area of pressure for a stroke of the first piston. In the second cylinder, the area of pressure, *il*, measures the effective work for a stroke of the second piston.

It may now be shown how to find the pressure at intermediate points of the curve of expansion, *gh*, into the second cylinder; $a\mathrm{L}$ is the capacity of the first cylinder, and $\mathrm{R}'''a\mathrm{L}$ is the volume by expansion, when the part of the stroke s has been described. This volume consists of portions of the two cylinders, namely, of the first cylinder, the portion $a\,(\mathrm{L}-s)$, or the area multiplied by the part of the stroke remaining to be described; and, of the second cylinder, the portion $a's$, or the area multiplied by the part of the stroke already described. And,

$$\mathrm{R}'''a\mathrm{L} = a\,(\mathrm{L}-s) + a's = a\mathrm{L} + (a'-a)\,s.$$

Transposing,

$$(\mathrm{R}'''-1)\,a\mathrm{L} = (a'-a)\,s, \qquad (1)$$

and

$$s = \frac{(\mathrm{R}'''-1)\,a\mathrm{L}}{a'-a} \qquad (2)$$

That is to say, for a given ratio of expansion, the part of the stroke described is equal to the capacity of the first cylinder multiplied by the ratio of expansion minus 1, and divided by the difference of the areas of the cylinders.

The ratio of expansion, R''', for a given part of the stroke described, is deduced from the equation (2), thus: $\mathrm{R}'''-1 = \frac{(a'-a)\,s}{a\mathrm{L}}$; and,

$$\mathrm{R}''' = 1 + \frac{(a'-a)\,s}{a\mathrm{L}} \qquad (3)$$

That is to say, the ratio of expansion is equal to the product of the difference of areas and the part of the stroke described, divided by the capacity of the first cylinder, 1 being added to the quotient.

The pressure, P′, at any given point of the stroke is symbolically $P' = \frac{P}{R'''} = \frac{P}{1 + \frac{(a' - a)\,s}{a\,L}}$; and, multiplying the numerator and denominator by aL,

$$P' = P \times \frac{a\,L}{a\,L + (a' - a)\,s} \quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

That is to say, the pressure at any given point of the stroke of the second cylinder is found by multiplying the initial pressure by the capacity of the first cylinder, and dividing the product by the sum of this capacity plus the product of the difference of areas by the part of the stroke described.

NOTE.—Where there is clearance, or any other intermediate space between the two cylinders, it is to be reckoned in parts of the stroke and added to the values of L for the stroke and the initial capacity, and s as the measure of the expanded volume.

For example, taking the areas of the first and second cylinders as 1 and 3, the stroke as 1, and the initial pressure = 21 lbs., then $a = 1$, $a' = 3$, and $L = 1$. For half-stroke, $s = .5$, and the pressure at half-stroke of the second piston is, by formula (4),

$$P' = 21 \times \frac{1 \times 1}{(1 \times 1) + (3 - 1)\,.5} = 21 \times \frac{1}{1 + 1} = 10.5 \text{ lbs.}$$

Now, suppose the base *mn*, or the stroke of the piston *b*, fig. 166, to be divided into 10 equal parts, as in the annexed figure, 167, and that the initial pressure, *ng*, is 21 lbs. The values of *s*, or the parts of the stroke at the successive points of division, taking the stroke = 1, are

0, .1, .2, .3, .4, .5, .6, .7, .8, .9, 1.0;

and the pressures by expansion at each point of the stroke, supposing that there is not any clearance, are,

21, 17.5, 15, 13.1, 11.7, 10.5, 9.5, 8.7, 8.1, 7.5, 7 lbs. per sq. inch;

or, putting the initial pressure = 1, they are relatively as

1, .833, .714, .625, .555, .500, .455, .417, .385, 357, .333.

Let these relative pressures be set off successively on ordinates raised from the points 1, 2, 3, &c., the initial ordinate *ng* being taken as 1, whilst the final ordinate *mh* is .333, then the curve *gh* traced through the ends of these ordinates is the expansion-curve for the stroke of the second cylinder; and is also the curve of back-pressure on the first piston.

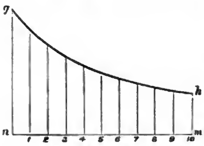


Fig. 167.
Ideal Expansion-curve of Steam.

On comparing the proportional pressures in the last line with those calculated for an ordinary hyperbolic curve of expansion,—the final ratios of expansion being the same in both cases, namely 3,—it is plain that the proportional pressures are the same in the two cases, varying from 1 to .333. The identity of the descending scales of pressure goes to show that the expansion-curve of the second cylinder, generated from the combined movements of two pistons, is hyperbolic in principle. Moreover, an analysis of the formula (4), by means of which the curve, fig. 167, was formed, proves that the one is the same in principle as the other.

III. WOOLF ENGINE, WITHOUT CLEARANCE—IDEAL DIAGRAMS.

The diagrams of pressure of the Woolf engine are produced in fig. 168, as they would be described by the indicator, according to the arrows. In these ideal diagrams, *pq* is the atmospheric line, *mn* the straight line of perfect vacuum, *cd* the straight line of admission; the terminal lines, *mc* and *ng*, straight and perpendicular to the atmospheric line, *dg* the hyperbolic curve of expansion in the first cylinder, and *gh* the consecutive expansion-line of back-pressure for the return-stroke of the first piston, and of positive pressure for the steam-stroke of the second piston. At the point *h*, at the end of the stroke of the second piston, the steam is exhausted into the condenser, and the pressure falls to the level of perfect vacuum, *mn*.

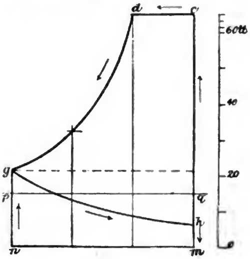


Fig. 168.
Woolf Engine:—Ideal Indicator Diagrams.

The diagram pertaining to the second cylinder, below the curve *gh*, is characterized by the absence of any specific period of admission; the whole of the steam-line *gh* being expansional, generated by the expansion of the initial body of steam contained in the first cylinder into the second. But the initial volume—the volume of the first cylinder—is gradually reduced by the advancing movement of the first piston, by which the initial steam is gradually driven into the second cylinder; until, when the stroke is completed, the whole of the steam is transferred from the first, and is shut into the second cylinder. The first cylinder, then, acts as a collapsible head to the second cylinder, for each steam-stroke of the latter; the second cylinder being, at the beginning of the stroke, augmented by the capacity of the first; and at the end of the stroke, reduced to its normal dimensions. The final pressure and volume of the steam in the second cylinder are, consequently, the same as if the whole of the initial steam had been admitted at once into the second cylinder, and then expanded to the end of the stroke in the manner of a single-cylinder engine; the hypothetical period of admission in the second cylinder being such a fraction of the stroke as would represent the ratio of the volume of the first cylinder, which is the volume of the steam admitted, to that of the second cylinder.

It may be added that the net work of the steam is also the same, according to both distributions. To prove this equality of work, let the diagram $ghmn$, fig. 169, be that of the second cylinder, and set off $n'n$, on the same base-line, mn, equal to one-third of the base-line, to represent proportionally the capacity of the first cylinder. Then the extended base-line $n'nm$ represents the augmented capacity of the second cylinder, at the beginning of the stroke; of which the augmentation $n'n$ represents the volume of the initial steam, of the pressure $n'g'$. The augmentation nn' is, as was said, gradually reduced by the advancing piston of the first cylinder, which may be conceived to move from the position $n'g'$ to the position ng, uniformly with the second piston in describing its stroke nm; so that, when the stroke is completed, the augmentation is eliminated, and the second cylinder returns to its normal capacity, measured by its base nm. Now, set off nn'' equal to one-third of the base-line nm, and draw the perpendicular $n''g''$; then nn'' represents the initial volume of the steam, or the period of admission, supposing that it had been admitted at once into the second cylinder, before expansion commenced, and $n''g''$ is the pressure when cut off. This volume would have expanded to three times itself, or the capacity of the second cylinder, and to the final pressure mh, which is one-third of the initial pressure $n''g''$ or ng; and the hyperbolic curve $g''h$ would have been the expansion-line. It is obvious that the two curves of expansion gh and $g''h$, the actual and the hypothetical, are identical, in so far that they have the same initial and the same final pressures, and that the intermediate pressures at proportional intervals are therefore, as has already been found, measured by ordinates of the same length. In so far, also, the curve of back-pressure, $g'h'$, exerted on the first piston whilst the second piston describes its stroke, is identical with the two others. The actual curves, gh and $g'h'$, may be defined as an elongation and a compression respectively of the true hyperbolic curve $g''h$.

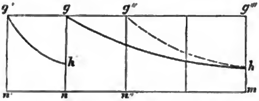


Fig. 169.—Woolf Engine:—Expansive Working.

From the identity of the three expansion-curves, gh, $g''h$, $g'h'$, in the sense just defined, it follows that the hyperbolic areas comprised between them and their bases are in the ratio of the lengths of base respectively, or as nm, $n''m$, $n'n$, which are as 3, 2, 1; and the rectangles over these bases respectively are also as 3, 2, 1. The differences, therefore, of the rectangles and the hyperbolic areas are likewise as 3, 2, 1; and further, the third difference, which is as 1, is equal to the difference of the first and second differences, since $3 - 2 = 1$. That is to say, the third difference, or the area $g'gh'$, is equal to the difference of the first and second differences or areas, $gg'''h$, and $g''g'''h$; that is, it is equal to the area $gg''h$ comprised between the first and second expansion-lines. Now, the area $g'gh'$ represents the uncounteracted useful work in the first cylinder, due to the fall of the back-pressure which accompanies the expansion into the second cylinder; and, according to the hypothetical distribution, it would be eliminated from the first cylinder, where the initial back-pressure would be maintained throughout the stroke, and replaced by the area $gg''h$, its equivalent of useful work, in the second cylinder; or, the work of the stroke of the second cylinder, measured by

the areas $ghmn + g'gk'$, according to the actual distribution,

is equal to the work measured by

the area $gg''hmn$, according to the hypothetical distribution.

The identity of net work thus proved is also shown clearly by constructing a combined diagram, with a continuous expansion-line, by piecing the first upon the second of the two diagrams, fig. 168, page 423, representing the work as if it were done in one cylinder equal in capacity to the second of the compound cylinders. For this purpose the first diagram is to be contracted to a length bearing the same proportion to the length of the second diagram that the volume of the first cylinder bears to that of the second cylinder—in this example, one-third. It is necessary to do so in order to reduce the two elements of the combined diagram to the same horizontal scale. When the strokes of the cylinders are equal to each other, the capacities are in the simple ratio of their areas. In this example, therefore, the length of the first diagram is to be reduced to one-third of the length of the second. For this purpose, let $ghmn$, fig. 170, annexed, be the diagram from the second cylinder, over which the original diagram from the second cylinder is shown in dotting. Draw gg''' parallel to the base, and mhc perpendicular to the base, equal to the height of the first diagram; set off $g'''g''$ equal to one-third of the base, and upon this reduced base $g'''g''$ complete the first diagram cdg'', by drawing cd parallel to the base, and equal to one-third of it, and the expansion-curve dg''. For the lower part of the figure, the line of back-pressure $g''h$ may be described by the method of ordinates, repeated from the curve gh. Thus the contracted diagram $cdg''h$ is completed. To combine the first diagram, thus reduced to uniformity of scale, with the second diagram, let again $ghmn$, fig. 171, annexed, be the second diagram, and describe the first diagram as reduced, in a

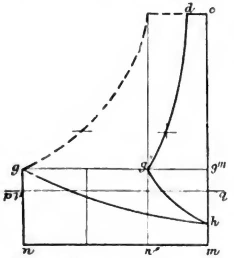


Fig. 170.—Woolf Engine:—Ideal Diagrams, reduced.

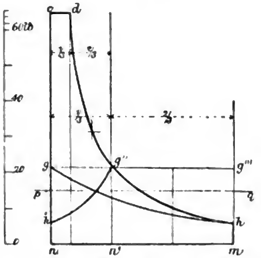


Fig. 171.—Woolf Engine:—Ideal Diagrams, combined.

reversed position *cdg″h′*, at the head of the second diagram, the same letters of reference being used. Finally, continue the hyperbolic expansion-line *dg″* to the end of the stroke at *h*. Then the area *gg″h*, is equal to the area *gg″h′*, and, when substituted for it, completes the regular indicator diagram *cdhmn*, with a continuous expansion-line *dg″h*. The substitution is, in fact, necessary, since the lower part of the first diagram partly overlaps the second diagram.

According to this combination, the upper part of the diagram, fig. 171, above the curve *gh*, namely, *cdhg*, represents the diagram as contracted, for the first cylinder: modified in form, but unaltered in area; and the lower part, below the curve *gh*, remains unaltered, both in form and in area, as the diagram for the second cylinder. The combined diagram, as a whole, exactly measures the whole net work done in both cylinders, and is such as would be formed by admitting and expanding the same quantity of steam in one cylinder having the dimensions of the second cylinder, with the period of admission, *cd*, equal to one-third of the capacity of the first cylinder, or one-ninth of the capacity or the stroke of the second cylinder.

It follows further, that the work effected by expansion into the second cylinder of the Woolf engine—that is, the total work arising from expansion against the second piston, plus the gain of work in the first cylinder by the gradual reduction of back-pressure in accordance with the expansion—is equal to that which would be effected by delivering the whole of the steam into the second cylinder before expansion is commenced, as in the receiver-engine. By this distribution, the upper part of the combined diagram, fig. 171, cut off by the horizontal line *gg″*, would measure the net work of the first cylinder, as there would be a uniform back-pressure equal to *ng* on the piston; and the lower part of the diagram, below *gg″*, would measure the work of the second cylinder, with a period of admission equal to *gg″*—the capacity of the first cylinder at the pressure *ng*—and with expansion to the end of the stroke.

To exemplify the foregoing conclusions, under the conditions originally stated, suppose that the steam is admitted to the first cylinder at a total initial pressure of 63 lbs. per square inch; that the areas of the first and second cylinders are respectively 1 and 3 square inches, and that the common length of stroke is 6 feet. The steam being cut off in the first cylinder at one-third of the stroke for the period of admission, *cd*, fig. 170, it is expanded to three times its initial volume, and to one-third of the initial pressure, namely, *ng*, equal to 21 lbs., at the end of the stroke. The steam is admitted to the second cylinder at the same pressure, *ng*, and is expanded there to three times the volume it acquired in the first cylinder, or to $3 \times 3 = 9$ times the initial volume in the first cylinder. At the same time, the pressure is reduced in the second cylinder to one-third of the final pressure in the first cylinder, or to one-ninth of the initial pressure there, namely, to 7 lbs. per square inch, measured by *mh*, fig. 170.

To calculate, from these data, the work for one stroke of the engine:—For the period of admission in the first cylinder, the total work is equal to

$mc \times cd = 63$ lbs. × 2 feet = 126 foot-pounds; and for the whole stroke of the first cylinder, the total work is,

$$126 \times (1 + \text{hyp log } 3) = 126 \times (1 + 1.0986) = 264.4236 \text{ foot-pounds.}$$

For the second cylinder, the work done is measured by the area of the diagram *ghmn*, fig. 170, and repeated in fig. 171. The area cannot be calculated directly on the hyperbolic principle, for the line of pressure, *gh*, is not a true hyperbolic curve; it is, as before stated, a hyperbolic curve elongated or stretched out, as shown in fig. 169 (p. 424), in which the actual curve *gh*, is an elongation of the hyperbolic curve $g''h$. Now, as the areas covered by these curves are proportional to their bases *mn* and mn'', or as 3 to 2, the first may, by means of this ratio, be found from the second, which is thus calculated on the hyperbolic rule:—The period of admission relative to the hyperbolic curve $g''h$ is nn'', which is one-third of the whole base *nm*; and the ratio of expansion is, therefore, 3. The initial pressure *ng* being 21 lbs. per square inch, the total initial pressure on the second piston is 21 lbs. × 3 square inches = 63 lbs.; and the work done during the hypothetical admission is

$$nn'' \times (ng \times 3) = 2 \text{ feet} \times 63 \text{ lbs.} = 126 \text{ foot-pounds.}$$

The work done during the hypothetical expansion, under the hyperbolic curve $g''h$, represented by the area $g''hmn''$, is, in terms of this initial work,—

$$126 \times \text{hyp log } 3 = 126 \times 1.0986 = 138.4236 \text{ foot-pounds;}$$

and the work done under the actual expansion-curve *gh* is found by increasing the last product in the ratio of the base mn'' to the base *mn*, thus,—

$$mn'' : mn :: 2 : 3 :: 138.4236 : 207.6354 \text{ foot-pounds,}$$

which is the work done for one stroke in the second cylinder.

The work of the back-pressure on the first piston, measured by the same diagram *ghmn*, is to be deducted from the positive work done on the second piston; and the remainder will be the net work done by expansion into the second cylinder. It is calculated by the same method as that employed for finding the positive work; and it is just one-third of the work on the second piston, or 69.2118 foot-pounds, since the two works must be in the ratio of the capacities of the first and second cylinders. Then, the net work by expansion into the second cylinder, is two-thirds of the total work; or it is,—

$$207.6354 - 69.2118 = 138.4236 \text{ foot-pounds.}$$

Adding together the total work of the first cylinder and the net work for the second cylinder, the sum is,—

$$264.4236 + 138.4236 = 402.8472 \text{ foot-pounds,}$$

for one stroke of the two cylinders.

It may be observed that the net work by expansion into the second cylinder is obtained by taking three-halves of the work by hypothetical expansion, and then deducting, for back-pressure on the first piston, one of these halves, showing that the work by the hypothetical expansion is equal to, and may be adopted as, the net work by the actual expansion,—so abridging the calculation.

But, the work of the compound engine may be calculated independently from the combined diagram, fig. 171, page 425; regarding the upper part of the figure, above the line gg'', as the net work of the first cylinder, according to the equivalent distribution mentioned at page 426, where the action is compared to that of a receiver-engine; and the lower part, below gg'', as the work of the second cylinder, with a period of admission equal to gg'', and an expansion to the end of the stroke. For the first section, the total work, over the base nn'', is calculated, and the work of the pressure, ng, as back-pressure, on the same base, is deducted from it to give the net work. Now, the total pressure, nc, calculated on the area of the second cylinder, is 63 lbs. × 3 square inches = 189 lbs.; and the period of admission, cd, is one-ninth of the stroke, or ⅔ foot. The total initial work is, then, 189 × ⅔ = 126 foot-pounds, and the total work, with an expansion of three times for the stroke gg'', 2 feet, is

$$126 \times (1 + \text{hyp log } 3) = 264.4236 \text{ foot-pounds.}$$

The work of the back-pressure, ng, or (21 × 3 =) 63 lbs., into the stroke, gg'', or 2 feet, is (63 lbs. × 2 feet =) 126 foot-pounds, and the net work, above the line gg'', is (264.4236 − 126 =) 138.4236 foot-pounds.

For the second section, according to the equivalent distribution, the initial work is equal to that of the back-pressure on the first piston, which has just been calculated, namely, 126 foot-pounds; and the work for an expansion of three times, through the stroke, nm, is found by what is only a repetition of the calculation for the upper section, to be 264.4236 foot-pounds.

The sum of the two sections is the total net work of the two cylinders; thus:—

Upper section, ...	138.4236	foot-pounds.
Lower section, ...	264.4236	„
Total net work, ...	402.8472	„

Otherwise, the combined diagram, fig. 171, represents the whole of the work as if it were done in one cylinder equal in capacity to the second cylinder—assumed, in this instance, to have the same diameter and stroke. The period of admission, cd, is one-ninth of the stroke, or (6 ÷ 9 =) ⅔ foot; the initial pressure being (63 × 3 =) 189 lbs., and the initial work (189 × ⅔ =) 126 foot-pounds. The whole work of the stroke is, therefore,

$$126 \times (1 + \text{hyp log } 9) = 126 \times 3.1972 = 402.8472 \text{ foot-pounds;}$$

the same as was calculated when the cylinders were taken separately.

IV. RECEIVER STEAM ENGINE, WITHOUT CLEARANCE.

The hypothetical distribution which has been described for the Woolf engine, according to which all the steam with which the second cylinder is charged, is supposed to be admitted into the second cylinder before expansion begins, is that which would take place in the ideal receiver-engine,—the second general combination of the compound engine,—in which the pistons of the two cylinders are connected to cranks at right angles to each other, on the same shaft, with an intermediate receiver. The receiver is occupied by steam exhausted from the first cylinder, and it supplies steam to the second cylinder, in which the steam is cut off and then expanded to the end of the stroke. On the assumption that the initial pressure in the second cylinder is equal to the final pressure in the first cylinder, and, of course, equal to the pressure in the receiver, the volume cut off in the second cylinder must be equal to the volume of the first cylinder, for the second cylinder must admit as much steam at each stroke as is discharged from the first cylinder.

For illustration, suppose, again, that the areas and capacities of the first and second cylinders, with the same length of stroke, are as 1 to 3, and that the steam is cut off at one-third of the stroke, and equally expanded in both cylinders, the ratio of expansion in each cylinder being thus equal to the ratio of the capacities of the cylinders. With this distribution, the volume admitted to the second cylinder is equal to the volume discharged from the first cylinder, and there is no intermediate fall of pressure.

Let A and B, fig. 172, be the first and second cylinders in section, in which the first piston *a* has arrived at the end of the down-stroke, and the second piston *b* has made half of its down-stroke; and let C be the intermediate receiver. The steam from the boiler is admitted to the first cylinder during the period *cd*, one-third of the stroke, when it is cut off and expanded to the end of the stroke, according to the expansion-curve *dg*, with a final pressure measured by *ng*. At the same time, steam of a pressure equal to *ng* has been exhausted from below the first piston into the receiver; and the steam now above the piston will be likewise exhausted during the return up-stroke, at the same constant pressure *ng*, into the reservoir, as indicated by the exhaust-line *gh*, parallel to the base of the diagram *mn*. In the second cylinder, steam from the receiver has been admitted at an initial pressure *oi*, equal to the final and exhaust pressure *ng* in the first

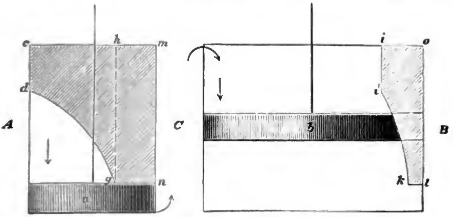


Fig. 172.—Receiver-Engine:—Ideal Action of Steam.

cylinder, and for a period of admission *ii'*, equal to that in the first cylinder, or one-third of the stroke; thence it is expanded to the end of the stroke, following the hyperbolic curve *ik*, with a final pressure *lk*.

V. RECEIVER-ENGINE, WITHOUT CLEARANCE—IDEAL DIAGRAMS.

Reproducing these ideal diagrams of pressure in fig. 173, in juxtaposition, draw *pq*, the atmospheric line. Then, *cd* is the line of admission, and *hg* the exhaust-line for the first cylinder, both of them being parallel to the atmospheric line; and *dg* is the expansion-curve. In the region below the exhaust-line of the first cylinder, between it and the line of perfect vacuum, the diagram of the second cylinder is formed; *hi*, the second line of admission, coincides with the exhaust-line *hg* of the first cylinder, and it is thus shown that, in the ideal diagrams, there is no intermediate fall of pressure. The line of perfect vacuum, *ol*, is parallel to the atmospheric-line, and *ik* is the expansion-curve. The arrows indicate the order in which the diagrams are formed.

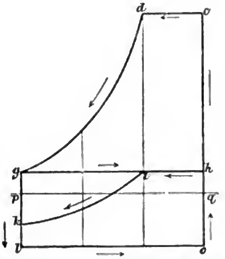


Fig. 173.—Receiver-Engine:—Ideal Indicator Diagrams.

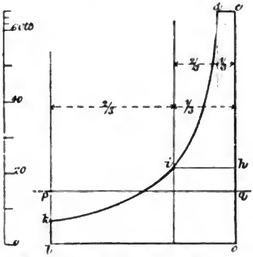


Fig. 174.—Receiver-Engine:—Ideal Diagrams, reduced and combined.

In the action of the receiver-engine, as above described and illustrated, the expansive working of the steam, though clearly divided into two consecutive stages, is, as in the Woolf engine, essentially continuous from the point of cut-off in the first cylinder to the end of the stroke of the second cylinder, where it is delivered to the condenser; and the first and second diagrams may be placed together and combined to form a continuous diagram. For this purpose, take, as was done for the Woolf engine, the second diagram as the basis of the combined diagram, namely, *h i k l o*, fig. 174, adding the atmospheric line *pq*. The period of admission, *hi*, is one-third of the stroke, and as the ratios of the cylinders are as 1 to 3, *hi* is also the proportional length of the first diagram as applied to the second. Produce *oh* upwards, and set off *oc* equal to the total height of the first diagram above the vacuum-line; and, upon the shortened base *hi*, and the height *hc*, complete the first diagram with the steam-line *cd*, and the expansion-line *di*.

By this construction, the regular indicator diagram $cdklo$ is formed, as applied to the second cylinder, and it measures the whole net work done in both cylinders:—the upper section, $cdih$, being the measure of the net work done in the first cylinder, and the lower section, $hiklo$, being the measure of the work of the second cylinder.

Resuming the data supplied for exemplifying the Woolf engine, with reference to the ideal diagrams for the receiver-engine, fig. 173, let the areas of the first and second cylinders be respectively 1 and 3 square inches, the stroke 6 feet, and the initial pressure in the first cylinder 63 lbs. per square inch. The steam being cut off at one-third of the stroke of the first cylinder, the final pressure is 21 lbs., and the total initial work therein is equal to $oc \times cd = 63$ lbs. × 2 feet = 126 foot-pounds; for the whole stroke the total work is

$$126 \times (1 + \text{hyp log } 3) = 126 \times 2.0986 = 264.4236 \text{ foot-pounds,}$$

the same as was found for the Woolf engine. For the second cylinder, the initial pressure is 21 lbs. × 3 square inches = 63 lbs., and the initial work is represented by $oh \times hi$, which is equal to (63 lbs. × 2 feet =) 126 foot-pounds. For the whole stroke, the total work is

$$126 \times (1 + \text{hyp log } 3) = 126 \times 2.0986 = 264.4236 \text{ foot-pounds.}$$

The work of the back-pressure, oh, on the first piston, which is continued for the whole of the stroke, is to be deducted from this total work; it is represented by the rectangle $oh \times hg$, equal to (21 lbs. × 6 feet =) 126 foot-pounds. Then (264.4236 − 126 =) 138.4236 foot-pounds is the net or effective work for the second cylinder for one stroke. This, the work for the second cylinder, is to be added to the work for the first cylinder, and the sum, 402.8472 foot-pounds, is the united work for one stroke of the two cylinders.

This is the same quantity of work as was calculated from the Woolf diagrams.

It is obvious that the two combined diagrams, figs. 171 and 174, pages 425 and 430, are identical in form and development.

The three calculations of work for one stroke of the two cylinders, thus made independently of each other, may be brought together for comparison as follows:—

Ideal Woolf Diagrams (fig. 170, page 425).

			foot-pounds.
First Cylinder:	Initial work, 63 lbs. × 2 feet = 126 foot-pounds.		
	Total work for whole stroke, 126 × (1 + hyp log 3)........		264.4236
Second Cylinder:	138.4236 × $^3/_2$ = ...	207.6354	
	Deduct for back-pressure on the first piston, one-third...	69.2118	
	Net work...		138.4236
	Work for one stroke of the two cylinders..................		402.8472

Ideal Receiver Diagrams (fig. 173, page 430).

			foot-pounds.
First Cylinder:	Initial work, 63 lbs. × 2 feet = 126 foot-pounds.		
	Total work for whole stroke, 126 × (1 + hyp log 3)........		264.4236
Second Cylinder:	Initial work, 21 lbs. × 3 square inches × 2 feet = 126 foot-pounds.		
	Total work for the whole stroke, 126 × (1 + hyp log 3) = ...	264.4236	
	Deduct for back pressure on the first piston, 21 lbs. × 6 feet = ...	126.0000	
	Net work..		138.4256
	Work for one stroke of the two cylinders..................		402.8472

Ideal Combined Diagrams (fig. 171, page 425, and fig. 174, page 430).

		foot-pounds.
Single Cylinder:	Equal in capacity to the second cylinder.	
	Initial work, 63 lbs. × 3 square inches × .666 foot ($^1/_9$ stroke) = 126 foot-pounds.	
	Work for the whole stroke, 126 × (1 + hyp log 9) = ...	402.8472

These calculations prove that, under the given conditions, when there is no intermediate fall of pressure, the performance of the steam is exactly the same, whether expanded direct from the first into the second cylinder, or through an intermediate receiver; and that the performance of both is identical with that of the same steam when expanded in and passed through one cylinder equal in capacity to the second cylinder; that, in fine, the performance of the steam is regulated simply by the initial pressure, and by the ratio of the total expansion behind a piston or pistons.

VI. WORK OF THE WOOLF ENGINE, WITH CLEARANCE.

The elementary principles on which the work of compound cylinders is to be estimated having been ascertained, it remains to introduce the element of clearance, and to find in what manner it modifies the conclusions above deduced from the distribution of steam without any clearance. For the purpose of this discussion, 7 per cent of clearance is assumed.

To begin with the Woolf engine, let the annexed fig. 175 represent the diagrams of pressure from the first and second cylinders of a Woolf engine having the same dimensions, proportions, and letters of reference, as in the preceding examples, with the addition of a clearance at each end of the first cylinder, measured by *cc'* or *mm'*, equal to 7 per cent of the stroke, or .42 foot. The rectangular clearance space *cc'm'm*, measures the passive work of the clearance, or the product of the pressure *mc* by the period of the clearance *cc.'*

As the steam is cut off at a third, or 33⅓ per cent, of the stroke, the actual ratio of expansion is $\frac{100+7}{33\frac{1}{3}+7}=2.653$. The initial pressure being 63 lbs., as before, the final pressure, *ng*, is $\frac{63}{2.653}=23.75$ lbs.; and the final volume, taking the working capacity of the first cylinder as 1, is 1 + (7 per cent), or 1.07. If there be no more clearance, or no intermediate space, between the first and second cylinders, the initial volume for expansion into the second cylinder is 1.07; and the final volume is 3. The ratio of expansion in it is, therefore, $\frac{3}{1.07} = 2.804$; and the final pressure, *mh*, is $\frac{23.75}{2.804} = 8.47$ lbs. per square inch.

In view of these pressures, it is apparent that, whilst the work of admission during the period *cd* is the same as it was when there was no clearance, namely, 126 foot-pounds, the work by expansion is greater, for the final pressures are respectively as follows:—

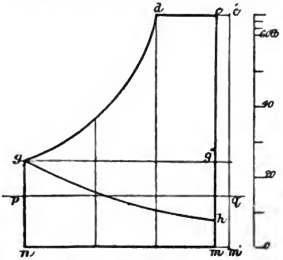


Fig. 175.—Woolf Engine:—Diagrams with Clearance.

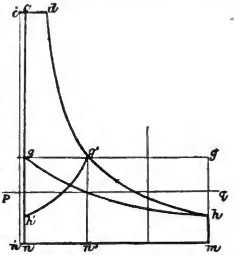


Fig. 176.—Woolf Engine:—The same diagrams reduced and combined.

	With no Clearance.	With Clearance.
In the first cylinder.........	21 lbs. per sq. inch.	23.75 lbs. per sq. inch.
In the second cylinder.....	7 „ „	8.47 „ „

To trace, as before, the equivalent distribution for the second cylinder, let the base *nm*, fig. 177, be the length of stroke or volume of the second cylinder; *n'n'''* the relative stroke or volume of the first cylinder, reduced for the same area as the second cylinder; and *n'''n* the clearance in the first cylinder, also relatively reduced. Then the base *n'n'''n* is the capacity plus the clearance of the first cylinder, or the initial volume for expansion in the second cylinder; and the base *n'''nm*, comprising the capacity of the second cylinder plus the clearance of the first cylinder, is the final volume by expansion. Set off *nn''*, one-third of the volume of the second cylinder, being equal to *n'n'''*, the capacity of the first cylinder, and draw the perpendicular *n''g''*; the volume *n'''nn''* is equal to the volume *n'n'''n*, and the hypothetical expansion-curve *g''h*, ending at the final pressure *mh*, would have been described if the steam

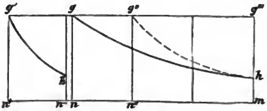


Fig. 177.—Woolf Engine:—Expansive Action.

had been admitted at once into the second cylinder, when the period of admission would have been gg''.

Following out the same argument as that which has already been applied to the diagram without clearance, the combined diagram, fig. 176, is readily constructed: forming a complete continuous diagram, *cdhmn*, which exactly measures the whole net work done in both cylinders, and is such as would be formed by admitting and expanding the same quantity of steam in a single cylinder having the dimensions of the second cylinder. But with this modification, that the clearance, measured by $c'c$, fig. 176, has the same volume as the clearance measured by cc', in the first cylinder, and has only a third of 7 per cent, or $2\frac{1}{3}$ per cent, of the stroke and the capacity of the second cylinder.

If, therefore, it were practicable to construct the compound engine so that there should not be any intermediate clearance, the employment of a smaller cylinder, as a prefix to a given cylinder, for receiving the charge of steam to be expanded through both cylinders, would have the effect of reducing the percentage of end-clearance, measured in parts of the larger cylinder. But it is not practicable to do so; and it remains to trace the influence of intermediate space combined with the initial clearance of the first cylinder, on the action and work of the steam in the second cylinder.

Take four cases, and suppose that the volume of the intermediate space, including what is technically the clearance of the second cylinder, is a simple fraction of the capacity of the first cylinder plus its clearance, 7 per cent, or of 1.07 times the capacity of the first cylinder, as follows:—

for the 1st, 2d, 3d, 4th case,
the intermediate spaces are,
0, $\frac{1}{3}$, $\frac{1}{2}$, 1, part of the capacity of the first cylinder plus its clearance; or they are,
0, ·357, ·535, 1.07 of the capacity of the first cylinder. Add to these 1.07, the capacity of the first cylinder plus its clearance; and the sums are the total initial volumes for expansion in the second cylinder,
1.07, 1.427, 1.605, 2.14, times the capacity of the first cylinder. Again, to the same values of the intermediate space, add 3, the capacity of the second cylinder; and the sums are the final volumes by expansion in the second cylinder,
3.0, 3.357, 3.535, 4.07 times the capacity of the first cylinder. The ratios of expansion in the second cylinder are the quotients of the final by the initial volumes:—

2.804, 2.352, 2.202, 1.902, ratios of expansion. The intermediate falls of pressure are, in parts of the final pressure in the first cylinder,
0, $\frac{1}{4}$, $\frac{1}{3}$, $\frac{1}{2}$ of the final pressure; or, putting the final pressure equal to 23.75 lbs., as was found, they are
0, 5.94, 7.92, 11.87 lbs. per square inch. The initial

pressures for expansion in the second cylinder are,

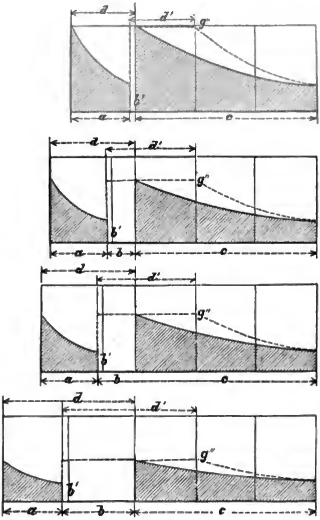
1, ¾, ⅔, ½ of the final pressure in the first cylinder; or

23.75, 17.81, 15.83, 11.87 lbs. per square inch; and the final pressures in the second cylinder are

8.47, 7.57, 7.19, 6.24 lbs. per square inch.

Here it is assumed, as before, that the intermediate space is unoccupied, or vacuous, when the exhaust is opened to it from the first cylinder.

The annexed diagrams, figs. 178, show the relative capacities of the cylinders with the clearance of the first cylinder and the intermediate space, and the hypothetical distribution in the second cylinder, for the four cases: a, the capacity of the first cylinder, with the curve of back-pressure; b, the total capacity of the intermediate space, including the clearance b'; c, the capacity of the second cylinder, three times that of the first; d, the total capacity of the first cylinder and its clearance; and d' its alternative or total initial volume for expansion in the second cylinder. The hypothetical points of cut-off g'', are at the same part of the stroke, one-third, in each case, and the final volumes for the second cylinder are equal to $(b+c)$ or e.



Figs. 178.
Woolf Engine:—Expansive Action, for the four cases.

Construct the combined diagrams, fig. 179, for the four cases, on the basis of the equivalent distribution above shown. The initial volume, $n'n''$, for expansion in the second cylinder, in the first case, consists of the net capacity of the first cylinder, nn'', augmented only by the initial clearance $n'n$; and the initial pressure is $n''g''$. For the second, third, and fourth cases, the respective initial pressures are $n''g^5$, $n''g^6$, $n''g^7$; and the falls of pressure are $g''g^5$, $g''g^6$, $g''g^7$, the corresponding initial volumes being $n''o'$, $n''o''$, $n''o'''$.

To examine more particularly, for example, the second case, it is to be remarked that the rectangle $g''g^5 \times g''c''$, is equal, in area, to the rectangle $n'c''' \times n'o'$; of which the former comprises the gain of work to the first cylinder under the segment gg'', by the fall of back pressure, together with the passive work in the clearance under the segment $c''g$; and the latter comprises the loss of initial work to the second cylinder by the intermediate space *n's*, with the space *so'* occupied by steam from the clearance. The equality of the rectangles may be proved by calculating their areas. For the first, the pressure $g''g^5$ is 5.94 lbs. per square inch; and 5.94 × 3 square inches, the area of the second cylinder, = 17.82 lb., which is the total fall of pressure. The length, $g''c''$, is 2.14 feet; and 17.82 lb. × 2.14 feet = 38.135 foot-pounds, which is the area of the first rectangle. For the loss, the pressure $n'c'''$ = 17.81 lbs. per square inch, and 17.81 × 3 inches = 53.43 lbs., the total pressure. The base $n'o'$ = .357 × 2 feet = .714 feet; and 53.43 lbs. × .714 = 38.149 foot-pounds, the area of the second rectangle, which, allowing for small errors in reduction, is the same as the area of the first rectangle. These rectangles represent the gross gain and the gross loss respectively, including the clearance; and, without following out the calculation, it is obvious that the net gain and net loss, excluding clearance, are equal. By similar calculations, it is shown that the gains and losses in question for the third and fourth cases, are also equal to each other, and therefore compensatory.

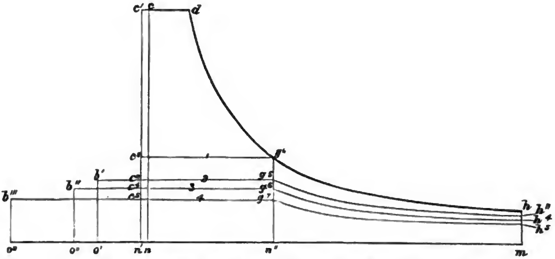


Fig. 179.—Woolf Engine, with Clearance:—Combined Diagrams, for the four cases.

The method of calculating the net work by means of a combination of the actual ratios of expansion in the two cylinders, without clearance, is applicable to compound cylinders as actually constructed,—with clearance.

The combined ratios in the four cases are as follows:—

			Combined Ratios.
1st case:—	1st ratio of expansion	1 to 2.653	
	2d do.	1 to 2.084	1 to 7.438
2d case:—	1st ratio of expansion	1 to 2.653	
	2d do.	1 to 2.352	1 to 6.241
3d case:—	1st ratio of expansion	1 to 2.653	
	2d do.	1 to 2.202	1 to 5.843
4th case:—	1st ratio of expansion	1 to 2.653	
	2d do.	1 to 1.902	1 to 5.046

The initial work of the steam of 63 lbs. total pressure, admitted into the first cylinder, for 2 feet of the stroke, and with a clearance of 7 per cent, or .42 foot, is as follows:—

Work done on the piston ...... 63 lbs. × 2 feet = 126 foot-pounds.
Work in the clearance........... 63 lbs. × .42 foot = 26.46 ,,

Total initial work of the steam, 63 lbs. × 2.42 feet = 152.46 ,,

This sum is the initial work on which the work by expansion is calculated; whilst it is 26.46 foot-pounds in excess of the initial work done on the piston. The total work is, then, calculated as follows:—

		Net Work in Foot-pounds.
1st case:—152.46 × (1 + hyp log 7.44) or 3.0069 =	458.27	
less, work in initial clearance	26.46	431.81
2d case:—152.46 × (1 + hyp log 6.24) or 2.8310 =	431.47	
less...	26.46	405.01
3d case:—152.46 × (1 + hyp log 5.84) or 2.7647 =	421.35	
less...	26.46	394.89
4th case:—152.46 × (1 + hyp log 5.05) or 2.6194 =	399.29	
less...	26.46	372.83

The calculations can be made more directly by means of the formula (3) at page 383. The reductions of net work, in the 2d, 3d, and 4th cases, are successively 6.2, 8.6, and 13.7 per cent of the work in the 1st case.

VII. WORK OF THE RECEIVER-ENGINE, WITH CLEARANCE.

For the work of receiver-engines, with clearance, taken at 7 per cent, at each end of the stroke of each cylinder, the annexed fig. 180 shows the diagrams of pressure, using the same data and letters of reference as in fig. 173, p. 430, with the clearance measured by *cc'*, *hh'*, or *oo'*, equal to

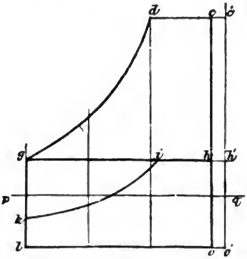


Fig. 180.—Receiver-engine:—Diagrams with Clearance.

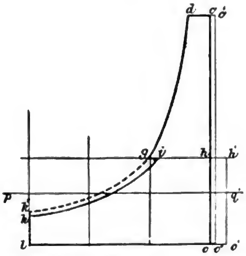


Fig. 181.—Receiver-engine:—The same diagrams reduced and combined.

7 per cent of *ol*. The steam being cut off at ⅓d, the actual ratio of expansion in the first cylinder is, as for the Woolf engine, $\frac{100+7}{33\frac{1}{3}+7}=2.653$; and the final pressure, *lg*, is $\frac{63}{2.653}=23.75$ lbs., which is also the pressure in the receiver, when there is no intermediate fall of pressure. The same is the initial pressure *oh*, in the second cylinder, with the clearance *oo′*. The volume admitted into the second cylinder is equal to the capacity of the first cylinder plus its clearance, or to one-third of the capacity of the second cylinder plus its clearance; that is, to one-third of 107 per cent, or 35⅔ per cent, which consists of the clearance, 7 per cent, and (35⅔ − 7 =) 28⅔ per cent of the stroke of the second cylinder. The steam admitted into the second cylinder thus occupies less than one-third of the stroke, by 4⅔ per cent, as indicated by the length of the period of admission, *hi*, in the diagram. As the steam is expanded from the capacity of the first cylinder plus its clearance, to that of the second cylinder plus its clearance, the ratio of expansion in the second cylinder, is necessarily equal to the ratio of the capacities of the two cylinders, which is 3; and $\frac{100+7}{28\frac{2}{3}+7}=3$; and the final pressure, *lk*, is $\frac{23.75}{3}=7.92$ lbs. per square inch.

The combined diagram, fig. 181, shows a dislocated expansion-line, in two parts: *dg* for the first cylinder, and *ik* for the second cylinder. The first part, *dg*, is extended continuously in dotting to the end of the stroke at *k′*, and shows the loss of work caused by the excess of the volume of clearance of the second cylinder over that for the first, as measured by the area of the strip *igk′k*.

For the other three cases, of intermediate falls of pressure, respectively ¼th, ⅓d, and ½ of the final pressure in the first cylinder. the relations are as follows:—

For the 1st, 2d, 3d, 4th case
the augmented initial volumes for expansion in the second cylinder are,
1, 1⅓, 1½, 2 times the capacity of the first cylinder plus the clearance; or they are
1.07, 1.427, 1.605, 2.14 times the capacity of the first cylinder. The final volumes by expansion in the second cylinder are equal to the capacity of the second cylinder plus its clearance, or to
3.21, 3.21, 3.21, 3.21 times the capacity of the first cylinder; and the ratios of expansion in the second cylinder are
3.00, 2.25, 2.00, 1.50.
The intermediate falls of pressure are
0, ¼, ⅓, ½ the final pressures in the first cylinder; and they are actually
0, 5.94, 7.92, 11.87 lbs. per square inch. The pressure in the reservoir, and the initial pressure in the second cylinder, are
23.75, 17.81, 15.83, 11.87 lbs. per square inch; and

the final pressures in the second cylinder are

7.92, 7.92, 7.92 7.92 lbs. per square inch.

The combined diagrams for these four cases are shown in fig. 182. The net losses of work, for the successive falls of intermediate pressure, are indicated by the small triangular areas ini', $in'i''$, $in''i'''$. Dividing the diagrams into three parts, for comparison; the upper part, $cdgh$, common to all the diagrams, is the normal net work for the first cylinder; the intermediate rectangles, hn, hn', hn'', are the accessions of work to the first cylinder by the fall of pressure; and the lower parts, below the lines of initial pressure, hi, $h''i'$, $h'''i''$, h^4i''', the works for the second cylinder.

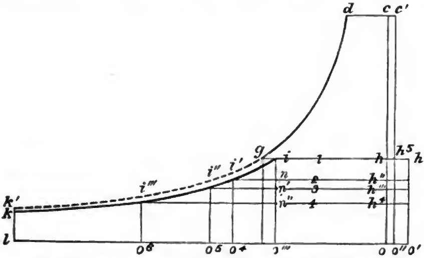


Fig. 182.—Receiver-engine:—Combined Diagrams, for the four cases.

To calculate the works done in the four cases:—First, the normal net work of the first cylinder, above the level of the terminal pressure, hig, common to all the cases. The total initial work done on the piston is, as was found before, 126 foot-pounds, and the work of the clearance 26.46 foot-pounds. The sum, 152.46 foot-pounds, is the initial work on which the work by expansion 2.653 times is calculated; thus:

	Foot-pounds.
$152.46 \times (1 + \text{hyp log } 2.653)$ or $1.9757 =$	301.21
Less work in initial clearance.........	26.46
Total work on piston, above the vacuum line..	274.75
Deduct work of back pressure, $oh \times hg$, or 23.75 foot-pounds × 3 inches × 2 feet	142.50
Normal net work on the first piston	132.25

Next, for the works gained to the first piston by the intermediate falls of pressure:—

For the	1st,	2d,	3d,	4th cases,
the work is expressed by the rectangles,				
	o,	$hh'' \times hg$,	$hh''' \times hg$,	$hh_4 \times hg$,
that is,	o,	5.94 lbs. × 3 × 2,	7.92 lbs. × 3 × 2,	11.87 lbs. × 3 × 2,
or,	o,	35.64,	47.52,	71.22 foot-pounds.

Third, for the work done in the second cylinder, the initial work for expansion must be the same as that for the first, namely, 152.46 foot-pounds; there being the same quantity of steam. The clearance is .42 foot, and the passive work in the clearance,

for the	1st,	2d,	3d,	4th case,
is	23.75 lbs. × 3 × .42,	17.81 × 3 × .42,	15.83 × 3 × .42,	11.87 × 3 × .42,
or	29.92,	22.44,	19.95,	14.95 foot-pounds.

The works in the second cylinder are calculated from these data, with the ratios of expansion, as follows:—

			Foot-pounds.
1st case:—	152.46 × (1 + hyp log 3) =	319.93	
	less the work in clearance	29.92	290.01
2d case:—	152.46 × (1 + hyp log 2.25) =	276.10	
	less the work in clearance	22.44	253.66
3d case:—	152.46 × (1 + hyp log 2) =	258.13	
	less the work in clearance	19.95	238.18
4th case:—	152.46 × (1 + hyp log 1.5) =	214.28	
	less the work in clearance	14.95	199.33

The total net work in both cylinders for one stroke, is found by adding together the three portions of work for each case:—

			Net Work in Foot-pounds.	Ratio of Net Work.
1st case:—	first cylinder above final pressure	132.25		
	intermediate	0.00		
		132.25		
	second cylinder	290.01	422.26, as	100
2d case:—	first cylinder above final pressure	132.25		
	intermediate	35.64		
		167.89		
	second cylinder	253.66	421.55, as	99.8
3d case:—	first cylinder above final pressure	132.25		
	intermediate	47.52		
		179.77		
	second cylinder	238.18	417.95, as	99.0
4th case:—	first cylinder above final pressure	132.25		
	intermediate	71.22		
		203.47		
	second cylinder	199.33	402.80, as	95.4

Here it is shown that the reduction of the quantity of work performed for one stroke of the pistons, by intermediate falls of pressure, is just

1 per cent when the fall amounts to even one-third of the final pressure in the first cylinder; and it is less than 5 per cent when the fall amounts to half the final pressure.

The work for one stroke may be calculated in terms of the combined ratios of expansion for the two cylinders; making allowances for the loss by clearance, and the gain to the first cylinder by the intermediate fall of pressure. The total initial work for expansion is, as was found, 152.46 foot-pounds; and the ratios of expansion are as follows:—

		Compound Ratio.
1st case:—first ratio of expansion..................	1 to 2.653	
second do.	1 to 3.000	1 to 7.959
2d case:—first ratio of expansion..................	1 to 2.653	
second do.	1 to 2.250	1 to 5.969
3d case:—first ratio of expansion..................	1 to 2.653	
second do.	1 to 2.000	1 to 5.306
4th case:—first ratio of expansion..................	1 to 2.653	
second do.	1 to 1.500	1 to 3.979

With respect to the first case, it is obvious from an inspection of the combined diagram, fig. 182, page 439, that the calculation of the work in terms of the initial work for expansion, and the total ratio of expansion, covers the whole area of the diagram, including the clearance-areas, thus:—

$$152.46 \times (1 + \text{hyp log } 7.959) \text{ or } 3.0743 = 468.71 \text{ foot-pounds.}$$

From this is to be deducted the work of the initial clearance in the first cylinder, 26.46 foot-pounds, represented by the area $cc'o''o$; and also the work of the excess of clearance in the second cylinder, over and above that of the clearance in the first cylinder, calculated on the pressure in the receiver. As the clearance of the second cylinder is, like that of the first, 7 per cent of the stroke, or .42 feet, the volumes of the respective clearances are in the ratio of the capacities of the cylinders, or as 3 to 1, and are measured by the spaces oo' and oo'' on the diagram. The clearance steam of the first cylinder, therefore, when transferred to the second cylinder, fills only one-third of its clearance space, measured by oo'', and of the pressure oh; and additional steam from the receiver, of the same pressure, is required to fill the remaining two-thirds of the clearance of the second cylinder, or $.42 \times \frac{2}{3} = .28$ foot, measured by $o''o'$. The work of the two clearances to be deducted is, then, as follows:—

Work of clearance of first cylinder, $cc'o''o$, 63 lbs. × 3 in. × .14 foot, or 63 lbs. × 1 in. × .42 foot....	= 26.46	foot-pounds.
Excess of clearance of second cylinder, $h'o'o''$, 23.75 lbs. × 3 in. × .28 foot........................	= 19.95	do.
Work of clearances........................	46.41	do.

The gross work of the diagram being, as above ....468.71 foot-pounds.
The work of clearances to be deducted is .... 46.41 do.

Net work for one stroke ....422.30 do.

For the 2d case, the gross work is—

152.46 × (1 + hyp log 5.969) or 2.7866 = 424.84 foot-pounds.
Deduct the work of the clearances, as above..... 49.41 do.

378.43 do.

To this is to be added the compensatory gain by the fall of the pressure in the reservoir, measured by hh'', which is equal to 5.94 lbs. per square inch. It is to be multiplied into the length, ghh', comprising gh, the stroke of the first cylinder, for the reduction of back pressure, and hh', the clearance of the second cylinder for the saving of passive work in the clearance. The stroke, gh, is, as reduced, 2 feet; the clearance, hh', is .42 foot, and the sum of these, gh', is 2.42 feet. Then the work of the gain is,

$hh'' \times gh'$ = 5.94 lbs. × 3 in. × 2.42 feet = 43.12 foot-pounds,
which is to be added to the remainder, above... 378.43 do.

making the net work for one stroke................421.55 do.

The calculations of net work are similarly performed for the 3d and 4th cases, and they are all brought together for the four cases for comparison, as follows:—

		Foot-pounds.
1st case:—152.46 × (1 + hyp log 7.959) or 3.0743	= 468.71	
deduct for clearances	46.41	422.30
2d case:—152.46 × (1 + hyp log 5.969) or 2.7866	= 424.84	
deduct for clearances	46.41	
	378.43	
add for fall of receiver-pressure 5.94 lbs × 3 in. × 2.42 feet	43.12	421.55
3d case:—152.46 × (1 + hyp log 5.306) or 2.6688	= 406.87	
deduct for clearances	46.41	
	360.46	
add 7.92 lbs. × 3 in. × 2.42 feet	57.50	417.96
4th case:—152.46 × (1 + hyp log 3.979) or 2.3810	= 363.01	
deduct for clearances	46.41	
	316.60	
add 11.87 lbs. × 3 in. × 2.42 feet..............	86.18	402.78

The net works thus arrived at are substantially the same as those that were deduced from the diagrams treated separately (page 440).

VIII. COMPARATIVE WORK OF STEAM IN THE WOOLF ENGINE AND THE RECEIVER-ENGINE.

It has been shown that the work of steam in the compound engine, when there is no clearance and no intermediate fall of pressure, is the same in amount, whether performed on the Woolf system or the receiver-system.

By the addition of clearance to each cylinder, equal to 7 per cent of the stroke at each end, the actual ratios of expansion are sensibly reduced as compared with the ratios without clearance. At the same time, it increases the net work done, as appears from the following statement:—

	Woolf Engine—7% Clearance.		Receiver-Engine—7% Clearance.	
	Ratio of Expansion.	Net Work.	Ratio of Expansion.	Net Work.
1st case:—	7.44	431.71 ft.-pds.	7.96	422.30 ft.-pds.
2d case:—	6.24	405.11 „	5.97	421.55 „
3d case:—	5.84	394.99 „	5.31	417.96 „
4th case:—	5.05	372.93 „	3.98	402.78 „

With clearance it is found that the reduction of net work, by intermediate fall of pressure, is less in the receiver-engine, where it is only 4½ per cent, with clearance, than in the Woolf engine, where it amounts to about 14 per cent, when the pressure falls intermediately one-half.

As the combined ratios of expansion in the receiver-engine are less, for each case, than in the Woolf engine; so the terminal pressures of the expanded steam in the second cylinder, on passing to the condenser, are greater in the receiver-engine than in the Woolf engine:—

For the 1st, 2d, 3d, 4th case, with 7 per cent clearance, the terminal pressures in the second cylinder are, for the Woolf engine,

8.47, 7.57, 7.19, 6.24 lbs. per square inch,

and for the receiver-engine,

7.92, 7.92, 7.92, 7.92.

In the first case, the terminal pressure in the Woolf engine is greater than in the receiver-engine; for there was no intermediate space assumed in the former, whilst clearance-space for the second cylinder was assumed in the latter; but, in the other cases, the terminal pressures in the former fall continuously below that of the first case. They also fall below those of the latter, which remain constant for all the cases. This constancy of terminal pressure in the second cylinder of the receiver-engine, simply follows from the fact that the terminal volume of the expanded steam is always the same,—that of the second cylinder plus the clearance,—whatever be the intermediate fall of pressure; whilst in the Woolf engine, on the contrary, the terminal volume is equal to that of the second cylinder, increased by the volume of the intermediate space, and the terminal pressure must be less as the terminal volume is increased.

As the terminal pressure in the receiver-engine is thus shown to be, in all practical cases, greater than in the Woolf engine, other conditions being the same, it directly follows that the work performed in expanding from a given initial pressure to the several terminal pressures, must be greater in the receiver-engine than in the Woolf engine.

It may be gathered from these arithmetical deductions that the receiver-engine is an elastic system of compound engine, in which considerable latitude is afforded for adapting the pressure in the receiver to the demands of the second cylinder, without considerably diminishing the effective work of the engine. In the Woolf engine, on the contrary, it is clearly of much importance that the intermediate volume of space between the first and second cylinders, which is the cause of an intermediate fall of pressure, should be reduced to the lowest practicable amount.

Supposing that there is no loss of steam in passing through the engine, by cooling and condensation, it is obvious that whatever steam passes through the first cylinder must also find its way through the second cylinder, neither more nor less. By varying, therefore, in the receiver-engine, the period of admission in the second cylinder, and thus also the volume of steam admitted for each stroke, the steam will be measured into it at a higher pressure and of a less bulk, or at a lower pressure and of a greater bulk: the pressure and density naturally adjusting themselves to the volume that the steam from the receiver is permitted to occupy in the second cylinder. With a sufficiently restricted admission, the pressure in the receiver may be maintained at the pressure of the steam as exhausted from the first cylinder. On the contrary, with a wider admission, the pressure in the receiver may fall or "drop" to three-fourths, or even one-half of the pressure of the exhaust-steam from the first cylinder.

There is a means of counterbalancing the loss of performance by intermediate fall of pressure, by so enlarging the second cylinder as to effect the same ultimate ratio of expansion behind the pistons, as would be effected in the originally designed engine if there were no intermediate fall. For example, when the capacities of the first and second cylinders are as 1 to 3, and the steam is cut off in each at one-third of the stroke, without any intermediate fall, the steam, if there be no clearance, is expanded into nine times its initial volume. But, when there is an intermediate fall of pressure, of, say, one-fourth of the final pressure in the first cylinder, involving an increase of volume of steam in the ratio of 3 to 4, the second cylinder must be correspondingly enlarged in the ratio of 3 to 4, in order to contain the charge of steam for expansion, when cut off, as before, at one-third of the stroke. By such enlargement of the second cylinder, in the ratio of the intermediate enlargement of the steam, the same ultimate ratio of expansion is secured, and an equivalent performance is effected. Such a remedy, when specially applied for the purpose of counterbalancing ineffective expansion of steam, involves the employment of enlarged cylinders, and entails the objections of increased weight, bulk, and cost of machinery. It would be more useful as a remedy, when applied to the Woolf engine, than to the receiver-engine.

IX. FORMULAS AND RULES FOR CALCULATING THE EXPANSION AND THE WORK OF STEAM IN COMPOUND ENGINES.

In view of the preceding discussions of the expansive working of steam in compound cylinders, the following algebraic symbols are used:—

a = the area of the first cylinder in square inches.
a' = the area of the second cylinder in square inches.
r = the ratio of the capacity of the second cylinder to that of the first cylinder.
L = the length of the stroke in feet, supposed to be the same for both cylinders.
l = the period of admission to the first cylinder in feet, excluding clearance.
c = the clearance at each end of the cylinders, in parts of the stroke, in feet.
L' = the length of the stroke plus the clearance, in feet.
l' = the period of admission plus the clearance, in feet.
s = the length of a given part of the stroke of the second cylinder, in feet.
P = the total initial pressure in the first cylinder, in pounds per square inch, supposed to be uniform during admission.
P' = the total pressure at the end of the given part of the stroke, s.
p = the average total pressure for the whole stroke.
R = the nominal ratio of expansion in the first cylinder, or $L \div l$.
R' = the actual ratio of expansion in the first cylinder, or $L' \div l'$.
R'' = the actual combined ratio of expansion behind the pistons, in the first and second cylinders together.
R''' = the actual ratio of expansion, or number of volumes into which the steam occupying the first cylinder at the end of the stroke, is expanded in the second cylinder at the end of any part of the return-stroke, s:—the special initial volume, or the capacity of the first cylinder, being = 1.
n = the ratio of the final pressure in the first cylinder to any intermediate fall of pressure between the first and second cylinders.
N = the ratio of the volume of the intermediate space in the Woolf engine, reckoned up to, and including the clearance of, the second piston, to the capacity of the first cylinder plus its clearance.[1]
w = the whole net work in one stroke, in foot-pounds.

Formulas and rules may be constructed on the basis of the combined ratio of expansion behind the two pistons:—the combined ratio being the product of the actual ratios of expansion in the first and the second cylinders. When, as usually happens in practice, intermediate expansion takes place between the cylinders, if the ratio of this expansion be multiplied into the combined ratio of expansion behind the pistons; or, if the three individual ratios of expansion in the first and second cylinders, and the intermediate space, be multiplied together, the product is the ratio of total expansion of the steam within the engine, to the end of the stroke of the second cylinder, when it is discharged into the condenser. For example, if the steam be expanded to three times its volume in the first cylinder, twice in the second cylinder, and one-and-a-half times in the intermediate space; the combined ratio of expansion behind the pistons is the product of the first and second of these; that is,—

---

[1] It is to be explained with respect to the value of N, that it is correctly expressed by the actual ratio of the volumes above stated, on the assumption that the intermediate space is a vacuum when it receives the exhaust-steam from the first cylinder. In point of fact, there is a residuum of unexhausted steam in the intermediate space, at low pressure, and the value of N is thereby practically reduced below the ratio above stated.

In first cylinder	1 to 3,	or 3
In second cylinder	1 to 2,	or 2
Combined ratio of expansion behind pistons	1 to 6,	or 6
Intermediate expansion	1 to 1.5,	or 1.5
Ratio of total expansion	1 to 9,	or 9

Or, the individual ratios may be placed consecutively thus:—

In first cylinder	1 to 3,	or 3
Intermediate expansion	1 to 1.5,	or 1.5
In second cylinder	1 to 2,	or 2
Ratio of total expansion	1 to 9,	or 9

Conversely, when the total expansion is given, the expansion behind the pistons may be calculated by dividing the total ratio by the ratio of intermediate expansion. Thus, if the total ratio be 9, and the intermediate ratio be 1.5, the combined ratio behind the pistons is $\frac{9}{1.5}=6$; or 1 to 6. Generally, if the total ratio be divided by any one of the individual ratios, the quotient is the product or combination of the two others.

Further, if two ratios be equal to each other, the combined ratio is equal to the square of one of them; and, conversely, the square root of a given ratio is the value of two elementary ratios, which when combined yield the given ratio. Thus, if there be two ratios, each equal to 3, then 3×3, or $3^2=9$, the combined ratio formed by those two. Conversely, ($\sqrt{9}=$) 3 is the value of two elementary ratios which, if combined, form the ratio 9. Similarly, the two equal ratios which, when combined, form the ratio 7, are each equal to $\sqrt{7}=2.65$; the square of 2.65, or 2.65×2.65, being equal to 7.

To find the Actual Ratio of Expansion in the first cylinder.—This is found by the formula (*a*), page 382, when the stroke, the period of admission, and the clearance are given. It is equal to

$$\frac{L'}{l'}=R' \qquad (5)$$

That is to say, the actual ratio of expansion in the first cylinder is equal to the quotient of the length of stroke plus the clearance divided by the period of admission plus the clearance. For example, if the steam be cut off at one-third of the stroke, and the clearance be 7 per cent, the length of stroke being equal to 1; then the stroke plus the clearance is equal to 1.07, and the period of admission is equal to $.3333+.07=.4033$; and $\frac{1.07}{.4033}=2.653$, the actual ratio of expansion.

To find the Ratio of Intermediate Expansion.—According to the assumption that the volume of a given weight of steam is inversely as its elasticity, or its pressure per square inch, the enlargement of volume, or expansion, may be deduced from the pressures before and after intermediate expansion. Thus, if the pressure be reduced from 20 lbs. to 15 lbs. per square inch, the volume, inversely, is enlarged in the ratio of 15 to 20; and, if the initial volume be taken as 1, then, by proportion, $15 : 20 :: 1 : 1.33$; or $1 \times \frac{20}{15} = 1.33$. Thus, in reducing the pressure $\frac{1}{4}$th, the volume is enlarged $\frac{1}{3}$d, and the ratio of expansion is 1.33.

By the notation, n is the ratio of the final pressure in the first cylinder, to the intermediate fall of pressure between the first and second cylinders; or, it is the denominator of the fractional part of the final pressure, expressing the fall of pressure. When the fall is $\frac{1}{4}$th, therefore, $n=4$; and the remaining pressure is $\frac{3}{4}$ths, and is as 3, or $n-1$. The pressures, then, before and after the fall, are as $n : n-1$; and, inversely, the volumes are as $n-1 : n$. Taking the capacity of the first cylinder with its clearance, as 1, the expanded volume is found by the proportion $n-1 : n :: 1 : \frac{n}{n-1}$; and the ratio of intermediate expansion is equal to

$$\frac{n}{n-1} \dots\dots\dots\dots\dots\dots \quad (a)$$

Substituting 4 for n in this expression, the ratio of expansion in the preceding example, is $\frac{4}{4-1} = \frac{4}{3} = 1.33$, as already found.

It is necessary, in the receiver-engine, thus to reckon backwards,—from the observed pressures to the volumes,—in order to find the intermediate ratio of expansion, since the volume of the receiver affords no evidence whatever of the amount of expansion between the first and second cylinders.

For the Woolf engine, the same process may be employed, to find the actual intermediate expansion. Otherwise, it is necessary to make an allowance off the intermediate space for the residual steam left in it about the end of each stroke of the second cylinder, before the new steam from the first cylinder is exhausted into the space. The residual steam is compressed by the new steam from the first cylinder, up to the pressure to which this steam falls by expansion into the intermediate space; and the total volume of this space is occupied by the two steams jointly,—distributed between them in the ratio of their weights. The intermediate volume from which is to be reckoned the intermediate expansion from the first cylinder, is therefore equal to the total intermediate space minus the space occupied by the compressed residuum of steam. To calculate the space thus occupied, it is only needed to multiply the total volume of intermediate space by the absolute pressure of the residual steam, and divide the product by the absolute pressure when the steam is exhausted from the first cylinder, since the volume of steam is, for this purpose, inversely as the absolute pressure; and the remainder of intermediate space left after deducting this fraction, is the intermediate volume to be adopted in the calculation of actual ratios of expansion.

The ratio of the capacity of the first cylinder plus the clearance, to the corrected intermediate space is, by the rotation, as 1 to N; and the sum of these, or the enlarged volume, is as $(1+N)$. The ratio of intermediate expansion is, therefore, as 1 to $(1+N)$; or it is

$$(1+N) \text{.......................................} \quad (b)$$

To find the value of N in terms of the intermediate fall of pressure:—The intermediate ratio was found, in terms of the ratio of pressure n, to be $\frac{n}{n-1}$; and $1+N=\frac{n}{n-1}$; so that

$$N=\frac{n}{n-1}-1 \text{.......................................} \quad (6)$$

That is, the corrected volume of the intermediate space relative to that of the first cylinder plus its clearance, is equal to the quotient of the final pressure in the first cylinder, divided by the reduced pressure after the fall, minus 1. For example, the final and reduced pressures being 20 lbs. and 15 lbs. respectively, $20 \div 15 = 1.33$; and $1.33-1=.33$, which is the corrected valve of N, the intermediate space, relative to the capacity of the first cylinder plus its clearance, taken as 1. In this calculation, the actual values of the pressures have been used, instead of their relative values as indicated in the above equation (6); but the result is the same, for, putting for n the ratio 4, then, $\frac{4}{4-1}=\frac{4}{3}=1.33$; and $1.33-1=.33$, as before.

The corrected capacity of the intermediate space in the Woolf engine is found by multiplying that of the first cylinder plus its clearance by the corrected ratio N.

To find the Ratio of Expansion in the second cylinder.—In the Woolf engine. This would be expressed by the ratio of the capacity of the first cylinder to that of the second cylinder, if there were no clearances nor other intermediate space. With clearances and intermediate space, the ratio of expansion in the second cylinder is less than that, and is equal to the ratio of the capacity of the first cylinder plus its clearance plus the corrected intermediate space, to the capacity of the second cylinder plus the corrected intermediate space, this last being taken to include the clearance of the second cylinder. Taking the capacity of the first cylinder plus its clearance as 1, that of the corrected intermediate space is N. The capacity of the second cylinder, with its clearance, is expressed by the ratio r; without clearance, it is less than r by as much in proportion as the capacity of the cylinder is less than the cylinder plus the clearance, or as L is less than $(L+c)$, or L′. The reduced ratio is, then, $r \times \frac{L}{L'}$; and the ratio of expansion in the second cylinder is as $(1+N)$ is to $\left(r \times \frac{L}{L'}\right)+N$; or

$$\text{Ratio of expansion in second cylinder} = \frac{\left(r \times \frac{L}{L'}\right)+N}{1+N} \text{..............} \quad (7)$$

That is, the ratio of the first to the second cylinder is multiplied by the length of stroke, and divided by this length plus the clearance; and the ratio of the corrected intermediate space is added to the quotient, making a sum, say, A. To the ratio of the corrected intermediate space is added 1, making a sum, say, B. Sum A is divided by sum B, and the quotient is the ratio of expansion in the second cylinder. For example, let $r=3$, $N=.333$, $L=6$ feet, and $L'=6$ plus 7 per cent of 6, or 6.42. Then

$$\frac{\left(3 \times \frac{6}{6.42}\right) + .333}{1 + .333} = 2.353,$$

the ratio of expansion in the second cylinder.

In the Receiver-engine.—The actual ratio of expansion, in the second cylinder, is not affected by clearance, assuming, of course, that the percentage of clearance is the same as in the first cylinder. When there is no intermediate fall of pressure, the ratio of expansion is simply that of the first and second cylinders, or r. But with an intermediate fall, this ratio is reduced as the ratio of intermediate expansion is increased, namely, $\frac{n}{n-1}$; and it is as this ratio inversely, or,

$$\text{Ratio of expansion in second cylinder} = r \times \frac{n-1}{n} = \frac{(n-1)\,r}{n} \text{ .... } (8)$$

For example, putting the ratio of the cylinders, $r=3$, and the ratio, n, of the final pressure in the first cylinder to the intermediate fall$=4$, as before; then, $\frac{(4-1)\,3}{4} = \frac{3 \times 3}{4} = 2.25$, the actual ratio of expansion in the second cylinder.

To find the Total Actual Ratio of Expansion as well as the Combined Actual Ratio of Expansion behind the two pistons.—The total actual ratio of expansion is, as was stated (page 445), the product of the ratios of the three consecutive expansions:—in the first cylinder, in the intermediate space, and in the second cylinder.

For the Woolf engine.—The expressions of these expansions are numbered (5), (*b*), and (7), and their product is

$$\frac{L'}{l} \times (1 + N) \times \frac{\left(r \times \frac{L}{L'}\right) + N}{1 + N} = \frac{L'}{l} \times \left(r \frac{L}{L'} \times N\right).$$

$$\text{Total actual ratio of expansion} = R' \left(r \frac{L}{L'} + N\right). \text{ .... } (9)$$

That is to say, the ratio, r, of the first to the second cylinder is to be multiplied by the length of stroke, and divided by this length plus the clearance; and the corrected ratio-value of the intermediate space, N, is added to the quotient. The sum is then multiplied by the actual ratio of expansion in the first cylinder, and the product is the total actual ratio of expansion. For example, let the steam be cut off at a third of the stroke of the first cylinder, with a clearance of 7 per cent; let the ratio,

r, of the cylinders be 3, and the corrected ratio-value, N, of the intermediate space, .333 or ⅓d. Then, the stroke of the first cylinder being = 1, the actual ratio of expansion in it, R′, as was exemplified at page 446, is $1.07 \div .4033 = 2.653$. The modified ratio of the cylinders is $3 \times \frac{1}{1.07} = 2.804$; and $2.804 + .333 = 3.137$. Finally, $2.653 \times 3.137 = 8.322$, the total actual ratio of expansion. It may be observed, that the fraction, $\frac{1}{1.07}$, above employed, is equivalent to the fraction, $\frac{6}{6.42}$, employed for the same purpose, in the example, page 449.

The combined actual ratio of expansion behind the pistons, in the Woolf engine, is the product of the first and third of the above-cited expressions, namely (5) and (7), or $\frac{L'}{l'} \times \frac{\left(r \times \frac{L}{L'}\right) + N}{1 + N}$; and, by reduction,

$$\text{Combined ratio of expansion on pistons} = \frac{R'\left(r \frac{L}{L'}\right) + N}{1 + N} \text{.......} \quad (10)$$

That is to say, the product, as above found (9), for the total actual expansion, is to be divided by the ratio value of the corrected intermediate space plus 1; the quotient is the combined actual ratio of expansion behind the pistons. For example, resuming the data of the preceding example, the final product expressing the total actual ratio of expansion, was found to be 8.322; and the divisor to be applied to it, is $1 + .333 = 1.333$. Then, $\frac{8.322}{1.333} = 6.242$, the combined actual ratio of expansion behind the pistons.

For the Receiver-engine.—The total actual ratio of expansion is the product of the expressions of the three consecutive expansions, numbered (5), (*a*), and (8); or, $\frac{L'}{l'} \times \frac{n}{n-1} \times \frac{(n-1)\,r}{n}$; and, by reduction,

$$\text{Total actual ratio of expansion} = r\frac{L'}{l'} = r\text{R}' \text{.................} \quad (11)$$

That is to say, the ratio, r, of the first and second cylinders, is to be multiplied by the actual ratio of expansion, R′, in the first cylinder. The product is the total actual ratio of expansion. For example, making as before, $r = 3$, and $\text{R}' = 2.653$, the product $(3 \times 2.653 =)$ 7.959, is the total actual ratio of expansion.

The product of the first and third of the above three expressions, namely (5) and (8), gives the value of the combined actual ratios of expansion behind the pistons; thus, $\frac{L'}{l'} \times \frac{(n-1)\,r}{n}$; and, by reduction,

$$\text{Combined ratio of expansion on pistons} = \frac{n-1}{n} r\text{R}' \text{........ ..} \quad (12)$$

That is to say, the ratio of the first and second cylinders is multiplied by the actual ratio of expansion in the first cylinder, and by the ratio of the intermediate fall of pressure to the final pressure in the cylinder minus 1;

and the final product is divided by this ratio simply. The quotient is the combined actual ratio of expansions behind the pistons. For example, resuming the product in the last preceding example, and taking n, the ratio of the intermediate fall of pressure $=4$; then $3 \times 2.653 \times \frac{4-1}{4} = 7.959 \times \frac{3}{4} = 5.969$, the required ratio.

To find the Work done in the two cylinders of compound engines—The Woolf engine.—It has already been stated that the formula (1), page 383, for the work of steam expanded in one cylinder, applies also to the work of steam in the Woolf engine, when the combined actual ratio of expansion behind the pistons in the two cylinders, is given. Thus, the total work for one stroke of the two pistons, quoting that formula, is,

$$w = aP\,[l'\,(1 + \text{hyp log } R'') - c] \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (13)$$

RULE 1. *To find the work done by steam in the two cylinders of a Woolf engine, for one stroke, with a given cut-off, and a given combined actual ratio of expansion.*—To the hyperbolic logarithm of the combined actual ratio of expansion behind the two pistons, add 1; multiply the sum by the period of admission to the first cylinder plus the clearance, in feet; and from the product subtract the clearance. Multiply the area of the first piston, in square inches, by the initial pressure in pounds per square inch, and by this remainder. The product is the total work in foot-pounds.

For example, let the 2d case, pages 436, 437, be calculated by this rule:—$aP = 1 \times 63 = 63$ lbs., $l' = 2.42$ feet, $c = .42$ foot, and $R'' = 6.24$. Then,

$$\begin{aligned} w &= 63\,[2.42\,(1 + \text{hyp log } 6.24) - .42] \\ &= 63\,[\,(2.42 \times 2.8310) - .42] \\ &= 63\,(6.851 - .42) = 405.20 \text{ foot-pounds,} \end{aligned}$$

as was before calculated, allowing for small errors of approximation.

The Receiver-engine.—A complete formula for the work of the receiver-engine necessarily comprises three elements:—First, the expression of the gross work, including the passive work of the clearances; second, the deduction for the passive work of the clearances; third, the addition for the gain of work by the reduction of the back-pressure on the first piston when there is an intermediate fall of pressure. Beginning with the first case, pages 441, 442, in which there is no intermediate fall of pressure, the total initial work of the steam admitted to the first cylinder is expressed by aPl'; whence the total work with expansion is

$$aPl'\,(1 + \text{hyp log } R'') \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (c)$$

This measures the total area of the diagram, fig. 182, page 439, including the clearances. The work of the clearance of the first cylinder, $cc'o''o$, is aPc. The work of the clearance of the second cylinder is the rectangle $hh'o'o$, which includes the section hoo'' of the first clearance; and, deducting this, there is left the remainder, the rectangle $h'o'o''$, to be added to the first clearance. To express this remainder algebraically, the volumes of the first and second clearances, oo'' and oo', are in the ratio of the areas of the cylinders, or as 1 to r, and the volume of the difference, $o''o'$, is as $c\,(r-1)$. The height, $o'h'$, is the final pressure in the first cylinder, and is equal to the initial pressure divided by R′, the actual ratio of expansion in the first cylinder, or $\frac{P}{R'}$. Therefore the work of the excess, $o''o'$, of the second clearance is $aPc\frac{r-1}{R'}$; and the two works of the clearances are together, $aPc\,(1+\frac{r-1}{R'})$, to be deducted from the gross work by expansion (c). Whence the equation for the total work, in the first case:—

$$w = aPl\,(1+\text{hyp log } R'') - aPc\left(1+\frac{r-1}{R'}\right);\ \text{or}$$

$$w = aP\left[l\,(1+\text{hyp log } R'') - c\left(1+\frac{r-1}{R'}\right)\right] \quad\ldots\ldots\ldots\ldots\ldots \quad (14)$$

when there is no intermediate fall of pressure.

Before reducing this formula to a rule, it may be remarked that it gives values which approximate closely to the true values, for cases in which there are intermediate falls of pressure—such cases as usually occur in practice;—and, for ordinary practical purposes, the results of the application of this formula will be sufficiently near to exactness. It was shown, in fact (pages 440, 441), that the reductions of work by intermediate falls, as compared with the work done when there is no fall, are as follows:—When the pressure falls to

¾, ⅔, ½ of the final pressure in the first cylinder,

the reduction of work is,

0.2, 1.0, 4.6 per cent of that in the first case.

The intermediate fall of pressure is rarely so much as two-thirds; and even with this fall, the reduction of work, it is seen, only amounts to 1 per cent. The slightness of the reduction results, as was before explained, from the fact that though the actual ratio of expansion, with intermediate falls, is less than when there is no intermediate fall, yet the loss of work by such reduction of expansion is practically compensated by the gain of net work on the first piston by the fall of the back pressure against it.

Adopting, then, the formula (14) as applicable for all cases of receiver-engines arising in practice, it is required only to give the actual ratio of expansion in the first cylinder, and to multiply this ratio by the ratio of the capacities of the two cylinders, to arrive at the ratio of expansion to be employed in the formula. This is literally the actual combined ratio of expansion for the first case, without intermediate fall of pressure, as was before found, represented by R″ in the formula (14).

RULE 2. *To find the total work done by steam in the two cylinders of a Receiver-engine for one stroke, with a given actual ratio of expansion in the first cylinder.*—Multiply the actual ratio of expansion in the first cylinder by the ratio of the two cylinders, and to the hyperbolic logarithm of the compound ratio so found add 1; multiply the sum by the initial period of admission to the first cylinder, plus the clearance, in feet (product A). Divide the ratio of the two cylinders, minus 1, by the actual ratio of expansion in the first cylinder; add 1 to the quotient, and multiply the sum by the initial clearance in feet (product B). Subtract product B from product A, giving the remainder C. Multiply the area of the first cylinder in square inches, by the total initial pressure in pounds per square inch, and by the remainder C. The product is the total work in foot-pounds for one stroke.

This rule is generally applicable not only for cases of no fall of pressure, but also for ordinary falls. $a=1$ square inch, $P=63$ lbs. per square inch, $c=.42$ foot, $l'=2.42$ feet, $R'=2.653$, $r=3$, $R''=7.959$, and hyp log $R''=2.0743$. Then, on the model of the given formula (14),

$$w=63\left[2.42\ (1+2.0743)-.42\left(1+\frac{3-1}{2.653}\right)\right]$$
$$=63\ (7.440-.737)=422.29 \text{ foot-pounds,}$$

as was before calculated for the first case. Or, following the wording of the rule:—The combined actual ratio of expansion is 7.959, of which the hyperbolic logarithm is 2.0743; adding 1 to this, the sum, 3.0743, is multiplied by 2.42, the initial period of admission plus the clearance, and $3.0743\times 2.42=7.440$ (product A). Again, the ratio of the cylinders is 3, and $3-1=2$; the actual ratio of expansion in the first cylinder is 2.653, and $2\div 2.653=.754$. Adding 1 to this quotient, the sum is multiplied by the initial clearance .42, or $1.754\times .42=.737$ (product B). The difference of products A and B is (7.440=.737=) 6.703, and this, multiplied by 63 lbs., the initial pressure per square inch, and by 1, the area of the piston in square inches, gives

$$6.703\times 63\times 1=422.29 \text{ foot-pounds,}$$

the work of one stroke.

If, nevertheless, it be desired to calculate with exactness the work for cases in which there are intermediate falls of pressure, the actual combined ratio of expansion is to be adopted, and allowance for the consequent gain of work on the first piston, is to be comprised in the formula. Referring to the diagram, fig. 182 (page 439), the gain by the fall of pressure was measured as a product of the fall hh'', by the stroke of the first cylinder plus its clearance as reduced, namely gh^5, plus the excess clearance of the second cylinder, namely, h^5h'; together, gh'. Now, the rectangle $gh^5\times h^5o''$, is the product of the expanded volume and pressure of the steam, and is equal to the initial work of the steam for expansion, or to aPl'; and the work of the excess, $o''o'$, of the second clearance, into the same pressure h^5o, was found to be equal to

$$aPc\frac{r-1}{R'}.$$

These two works together are expressed by

$$aPl'+aPc\frac{r-1}{R'};$$

and the gain by the intermediate fall of pressure, or $\frac{1}{n}$th of the final pressure $o'h'$, is $\frac{1}{n}$th of this sum, or

$$\left(aPl' + aPc\,\frac{r-1}{R'}\right)\times\frac{1}{n},$$

$$\text{or } aP\left[\left(l'\times\frac{1}{n}\right)+\left(c\,\frac{r-1}{R'}\times\frac{1}{n}\right)\right].$$

Adding this quantity to the value of w in equation (14), it becomes,

$$w = aP\left[l'\left(\left(1+\frac{1}{n}\right)+\text{hyp log } R''\right) - c\left(1+\frac{r-1}{R'}\right) + c\left(\frac{r-1}{R'}\times\frac{1}{n}\right)\right],$$

$$\text{or, } w = aP\left[l'\left(\frac{n+1}{n}+\text{hyp log } R''\right) - c\left(1+\frac{r-1}{R'}-\frac{r-1}{nR'}\right)\right];$$

and, by reduction, the general formula is arrived at:—

$$w = aP\left[l'\left(\frac{n+1}{n}+\text{hyp log } R''\right) - c\left(1+\frac{(n-1)\,(r-1)}{nR'}\right)\right] \ldots\ldots \quad (15)$$

Whence the following rule, which gives the exact work for one stroke, with an intermediate fall of pressure:—

RULE 3. *To find the exact total work done by steam in the two cylinders of a Receiver-engine, for one stroke, with any intermediate fall of pressure.*— To the denominator of the vulgar fraction which expresses the intermediate fall of pressure, add 1; and divide the sum by the denominator. To the quotient, add the hyperbolic logarithm of the combined actual ratio of expansion behind the two pistons; and multiply the sum by the initial period of admission to the first cylinder plus the clearance, in feet (product A). Multiply the said denominator minus 1, by the ratio of the two cylinders minus 1; also multiply the actual ratio of expansion in the first cylinder by the said denominator; and divide the first by the second of these products. Add 1 to the quotient, and multiply the sum by the initial clearance in feet (product B). Subtract product B from product A, giving the remainder C. Multiply the area of the first cylinder in square inches, by the total initial pressure in lbs. per square inch, and by the remainder C. The product is the total work in foot-pounds.

For example, calculate the second case of the receiver-engine, with an intermediate fall of ¼th the initial pressure, as given at pages 441, 442. The denominator of the fraction is 4, the combined actual ratio of expansion is 5.969, and the ratio for the first cylinder is, as before, 2.653. The hyperbolic logarithm of 5.969 is 1.7866, and

$$\frac{4+1}{4} + 1.7866 = 3.0366;\ \text{and } 3.0366 \times 2.42 \text{ ft.} = 7.3486 \ldots\ldots\ldots\ldots \text{(A)}$$

Again, $(4-1)\times(3-1) = 3\times 2 = 6$; and $2.653\times 4 = 10.612$; then

$$\left(1+\frac{6}{10.612}\right)\times .42 \text{ ft.} = .6574 \ldots\ldots\ldots\ldots \text{(B)}$$

$$\text{and } A - B, \text{ or } 7.3486 - .6574 = 6.6912 \ldots\ldots\ldots\ldots \text{(C)}$$

Finally, the area, 1 square inch × 63 lbs. × 6.6912 = 421.55 foot-pounds, the total work for one stroke, the same as was found before.

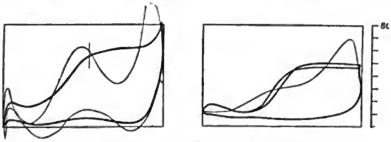
The above calculation is compactly expressed, on the model of the formula, thus,—

$$1 \text{ in.} \times 63 \text{ lbs.} \left[2.42 \text{ ft.} \left(\frac{5}{4} + \text{hyp log } 5.969 \right) - .42 \text{ ft.} \left(1 + \frac{3 \times 2}{4 \times 2.653} \right) \right] = 421.55 \text{ foot-pounds.}$$

The result arrived at by the application of the general rule given in pages 452 and 453, would be 422.29 foot-pounds, which is only ¾ foot-pounds in excess of the exact result just found.

CHAPTER IV.—BEHAVIOUR OF STEAM DURING ADMISSION INTO THE CYLINDER.

The undulatory movements of the indicator, at high speeds, so affect the conformation of the diagram, that its indications cannot always be accepted as literal transcripts of the varying pressures of the steam in the cylinder.[1] Nevertheless, the net area of the diagram, which is the measure of disposable power, ought to be correct in amount, as the action and reaction of elastic undulations are equal to each other. The dark-line diagram, fig. 183, was described by a locomotive (C. R., No. 42) running at a speed of 38 miles per hour, or 590 feet of piston per minute. The light-line diagram was caused to be described by suddenly opening the stop-cock communicating with the steam cylinder. The steam, by an instantaneous blow, had surprised the piston of the indicator into the extreme oscillations shown in light lining, throughout a complete double-stroke, after which the tracing of the pencil subsided into the constant figure in heavy lining. The net areas of the two figures are substantially the same in amount. Fig. 184 is a quieter example from another locomotive (E. & G. R., Pallas) running at 42 miles per hour, or 652 feet of piston per minute, in which case the diagrams were described in immediate succession. Two series of diagrams taken from long-stroked cylinder locomotives (C. R., Nos. 124, 125) at 31 miles per hour, or 640 feet of piston per minute, also exemplify the equality of areas already shown: the steam having been cut off at



C. R., No. 42—4th notch—38 m.p.h. E. and G. R., Pallas—10th notch—42 m.p.h.

Figs. 183, 184.—Undulatory Movements of the Indicator.

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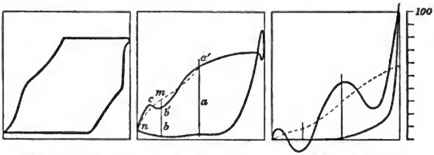
[1] See *Railway Machinery*, page 68, for a full discussion of the merits of the indicator.

54 per cent for the larger diagram, and at 75 per cent for the smaller diagram.

Undulatory diagrams may be reduced to the radical form, with the aid of the points of cut-off and exhaust. The diagram, fig. 185, was taken from the Brindley, E. & G. R., at a low speed; and it shows that the steam was cut off at 9.34 inches, and exhausted at 15 inches of the stroke of 18 inches. For the same points of cut-off and exhaust, the diagram, fig. 186, from the same engine was taken at 25 miles per hour, or 320 feet of piston per minute. Here, every angular deviation of the curve in fig. 185, is rounded in fig. 186. Draw perpendiculars a and b through the points of cut-off and exhaust. The indicated pressure at a' is no doubt the real pressure of the steam cut off. But the segment $a'b'$ must be replaced by a regular expansion curve $a'm$, and the segment bn by a regular exhaust-line mn, like those of fig. 185, excluding the rebounding prominence c; and such that the area inclosed by the new or normal lines of expansion and exhaust is equal to that of the literal diagram. Again, the diagram, fig. 187, from the Hebe, E. & G. R., at 48 miles per hour, or 680 feet of piston per minute, cutting off at 9 inches, and exhausting at 15½ inches, in a 20-inch stroke, is readily reducible on the same principles to the regular steam line in dotting.



E. & G. R., Brindley. Brindley, 25 m.p.h. E. & G. R., Hebe.

Figs. 185, 186, 187.—Undulatory Movements of the Indicator.

The second series of indicator diagrams, taken in 1850 and reduced by Sir Daniel Gooch, from the Great Britain, are given in one view in Plate I.[1] For the purpose of these indications, special trains were prepared for the Great Britain, and were run over a section of the Great Western Railway. They may be taken as standards of the performance of locomotives having well-protected inside-cylinders; for, though these diagrams were taken more than thirty-six years ago, the performance indicated by them has not been surpassed in locomotive practice. They were taken for three different periods of admission, for which the needful particulars are as follows:—

Great Britain.—Cylinders 18 inches in diameter, 24 inches stroke; driving-wheels, 8 feet in diameter; steam-ports, 13 inches by 2 inches; exhaust-port, 13 inches by 3½ inches; clearance, in parts of the stroke, 1.80 inches; steam-pipe, 4½ inches in diameter; blast-pipe, 5½ inches in diameter at orifice. Slide-valve—

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[1] These diagrams, with their order-numbers, are reproduced from the second part of Diagram-plate 4 of *Railway Machinery.*

lap, 1¼ inches; lead, ⅜ inch; maximum travel in 1st notch, 4¾ inches; travel in 3d notch, $3\frac{15}{16}$ inches; travel in 5th notch, $3\frac{7}{16}$ inches. Opening of steam-port for steam—1st notch, $1\frac{1}{32}$ inches; 3d notch, $\frac{21}{32}$ inch; 5th notch, ½ inch. Cut-off—1st notch, 16 inches, or 67 per cent of the stroke; 3d notch, 11¾ inches, or 49 per cent (taken by Sir Daniel Gooch at 12 inches, or 50 per cent); 5th notch, 7 inches or 29 per cent. Exhaust—1st notch, 21⅜ inches, or 89 per cent; 3d notch, 19¾ inches, or 82 per cent; 5th notch, 17⅜ inches, or 72½ per cent. Compression—1st notch, 3 inches; 3d notch, 5 inches; 5th notch, 7½ inches.

The following table, No. 97, contains dimensions and other particulars of the locomotives with which experiments have been made on the action of steam in the cylinder. The actual states of the valves at the times of the experiments are noted. The initials G. W. R. stand for the Great Western Railway; E. & G. R., the Edinburgh and Glasgow Railway; and C. R., the Caledonian Railway. All the engines were passenger locomotives, excepting C. R., Nos. 102, 124, 125, and 127, which were goods locomotives. Sections of the cylinders and smoke-boxes are shown in figs. 188, 189, and 190. Of the inside cylinders, which have been subjected to experiment, it may be explained that all of them were placed within the smoke-box, and, in general, totally surrounded by the atmosphere of hot smoke, although the covers were exposed. All the outside cylinders were those of locomotives on the Caledonian Railway. In general arrangement, the Caledonian passenger locomotives were much alike, the cylinders being fixed between outer and inner frame-plates, and embraced partially, at the upper part, by the plates of the smoke-box. At the outer and lower parts the cylinders were practically beyond the reach of the heat that existed in the smoke-box, and there was no felting about them. The cylinders of the goods engines, placed entirely clear of the smoke-box, were hung from the outside of the frame, not deriving any benefit from the warm atmosphere of the smoke-box, and protected from the weather only by a sheathing of felt and sheet-iron.

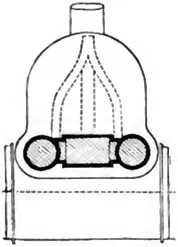


Fig. 188.—Great Britain, G. W. R.

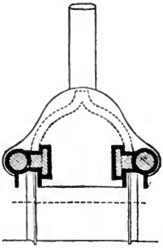


Fig. 189.—Passenger, C. R.

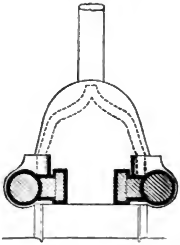


Fig. 190.—Goods, C. R.

Experimental Locomotives.

Table No. 97.—DIMENSIONS OF LOCOMOTIVES, RELATIVE TO THE ACTION OF STEAM IN THE CYLINDERS. 1850.

DESIGNATION OF LOCOMOTIVE.	Cylinder: diameter and stroke.	Driving-wheels: diam.	Steam-ports: length and width.	Area of steam-port: fraction of piston-area.	Slide-valve: full gear.			Position of Cylinders
					Lap.	Lead.	Travel.	
1	2	3	4	5	6	7	8	9
	inches.	ft. ins.	inches.		inches.	inches.	inches.	
G. W. R., Great Britain	18 × 24	8 0	2 × 13	1-10th	$1\frac{1}{4}$	$\frac{3}{8}$	$4\frac{3}{4}$	inside.
E. & G. R., Orion......	15 × 20	6 0	$1\frac{3}{16}$ × 10	1-14th	1	$\frac{1}{8}$	4	do.
" Hebe.......	15 × 20	5 6	$1\frac{3}{8}$ × $9\frac{1}{2}$	1-14th	$\frac{5}{8}$	0	$3\frac{1}{8}$	do.
" Nile........	16 × 18	6 0	$1\frac{1}{4}$ × $11\frac{5}{8}$	1-14th	1	$\frac{5}{16}$	$3\frac{3}{4}$	do.
" Pallas......	15 × 20	6 0	$1\frac{1}{4}$ × 10	1-14th	$\frac{3}{4}$	$\frac{1}{4}$	4	do.
" Brindley...	14 × 18	5 6	$1\frac{3}{8}$ × $6\frac{1}{2}$	1-17th	1	$\frac{1}{4}$	4	do.
C. R., No. 13............	15 × 20	6 0	$1\frac{3}{16}$ × 10	1-14th	$1\frac{3}{16}$	$\frac{5}{16}$	$4\frac{1}{2}$	outside
" No. 14............	15 × 20	6 0	$1\frac{3}{16}$ × 10	1-14th	$1\frac{3}{16}$	0	$4\frac{1}{4}$	do.
" No. 33............	15 × 20	6 0	$1\frac{1}{4}$ × 10	1-14th	$1\frac{1}{4}$	$\frac{1}{16}$	$4\frac{1}{2}$	do.
" Nos. 41, 42......	15 × 20	6 0	$1\frac{1}{4}$ × 10	1-14th	$1\frac{1}{4}$	$\frac{1}{4}$	$4\frac{3}{4}$	do.
" No. 51, old valve	15 × 20	6 0	$1\frac{1}{4}$ × 10	1-14th	$1\frac{1}{4}$	$\frac{1}{16}$	$4\frac{5}{8}$	do.
" No. 51, new "	15 × 20	6 0	$1\frac{1}{4}$ × 10	1-14th	$1\frac{1}{2}$	$\frac{1}{4}$	$5\frac{1}{4}$	do.
" No. 73............	13 × 18	5 0	1 × $8\frac{7}{8}$	1-15th	1	$\frac{5}{16}$	4	do.
" Nos. 124, 125...	17 × 24	4 7	$1\frac{1}{4}$ × 12	1-15th	$1\frac{1}{4}$	$\frac{5}{32}$	5	do.
" No. 127...........	17 × 24	4 7	$1\frac{1}{4}$ × 12	1-15th	$1\frac{1}{4}$	$\frac{1}{4}$	5	do.
" No. 102...........	16 × 18	4 6	$1\frac{1}{4}$ × $11\frac{1}{2}$	1-14th	1	0	4	do.

Wiredrawing.

In the following table, No. 98, are given the lowest speeds at which wiredrawing exists in the cylinders of the locomotives therein named, for any extent greater than 1 inch before the point of cut-off, and also the reduction of pressure by wiredrawing, at the highest observed speeds.

In each series, of the first part of this table, it is apparent that the shorter the admission the lower is the speed of piston at which wiredrawing becomes sensible; and, at the higher speeds, the greater is the fall of pressure by wiredrawing. This is explained by the reduction of travel and of maximum opening of port provided by the nature of the link-motion, for shorter admissions, and the consequent lower velocity of motion of the valve when cutting off steam. In the Great Britain, as may be noted in the figures in Plate I., wiredrawing is only developed when the speed of piston rises to from about 450 feet to 700 feet per minute, according as the admission rises from 29 per cent to 66 per cent of the stroke. As the steam-lines in the diagrams from this engine are, in general, straight from the beginning, until they near the cut-off, it is plain that, beyond a maximum opening of the port, in all cases much less than its total length, any extra extent of opening does not further promote the ingress of steam. The uniformity of pressure thus maintained, with a varying and limited opening of the port, up to the speeds at which wiredrawing prevails, even though the pressure in the valve-chest may be much greater than that in the cylinder, results from the frictional resistance to the movements of the steam along the steam-ways, which practically supersedes, at the lower speeds, the throttling action of the valve. As the speeds rise, the opening of the valve becomes insufficient for the maintenance of a constant pressure, since wiredrawing, and lowering of pressure, are set up at higher speeds, and increase with them: in consequence of the greater flow of steam through the port, by which a throttling action comes into play, augmented by the local acceleration of the speed of the piston in the first half of the stroke.

In the additional examples, in the following table, as in the previous cases, it is observable that the shorter the admission, the greater is the fall of pressure by wiredrawing in each cylinder, and that it also increases with the speed.

Table No. 98.—WIREDRAWING OF STEAM DURING ADMISSION TO THE CYLINDERS OF LOCOMOTIVES.

DESIGNATION OF LOCOMOTIVE.	Period of Admission.	Travel of Valve.	Port Opens.	Wiredrawing commences at Speed of		Wiredrawing at the Highest Speed attained.				
						Speed of		Pressure during Admission.		
				Engine.	Piston.	Engine.	Piston.	Initial.	Cut-off.	Difference.
	per cent.	inches.	inches.	miles per hour.	feet per minute.	miles per hour.	feet per minute.	lbs.	lbs.	per cent.
Great Britain	66	4¾	$1\frac{1}{32}$	49	686	54	756	89	86	3.5
Do.	49	$3\frac{15}{16}$	$\frac{21}{32}$	40	560	55	770	84	72	14
Do.	29	$3\frac{7}{16}$	½	31	434	56	784	90	65	28
C. R., No. 33.........	61	3½	½	36	558	40	620	36	34	6
Do.	55	3¼	⅜	30	465	41	635	62	59	5
Do.	35	$2\frac{13}{16}$	$\frac{5}{32}$	20	310	44	682	55	47	10
C. R., No. 125........	71	$4\frac{5}{16}$	⅞	22	540	29	711	27	25	7
Do.	55	3¾	⅝	14.5	355	30	735	64	53	17
Do.	35	3¼	⅜	15.5	380	26	637	54	40	26
Additional Examples, reduced from indicator diagrams.										
E. & G. R., Hebe...	75	3	⅞	33	561	40	680	34	25	26
Do.	58	—	—	13	221	41	697	50	26	48
Do.	45	—	—	—	—	30	510	72	48	33
Do.	45	—	—	—	—	45	745	52	23	57
E. & G. R., Orion...	57	3	$\frac{9}{16}$	20	310	30	465	58	48	17
Do.	50	2¾	$\frac{7}{16}$	18	279	40	620	56	39	30
E. & G. R., Nile....	68	3¾	⅞	30	465	40	560	55	45	18
C. R., No. 42.........	45	$3\frac{9}{16}$	$\frac{9}{16}$	—	—	38	589	70	56	20
Do.	18	3⅜	$\frac{7}{16}$	—	—	36	558	63	46	27
C. R., No. 13.........	62	4½	$1\frac{1}{16}$	—	—	54	837	58	49	15
C. R., No. 51 (old valve)............	41	$3\frac{11}{16}$	⅝	—	—	32	496	48	41	15
Do.	28	$3\frac{7}{16}$	½	—	—	31	480	62	55	11
C. R., No. 51 (new valve)............	46	4⅜	$\frac{13}{16}$	20	310	40	620	55	40	27
Do.	30	4⅜	$\frac{11}{16}$	20	310	49	760	45	35	22

Such of the engines, table No. 97, as have lost all, or nearly all, the lead of their valves, as specified in the table, show a less degree of wiredrawing than those which have sufficient lead. Thus, No. 33, which had lost nearly all lead, shows, for a shorter admission and a higher speed, a fall of 10 per cent of pressure, which is but half of that for No. 42, having a ¼-inch lead. No. 51, with the old valve and the new valve, shows a corresponding contrast; also No. 33 and No. 13. Such contrasts are explained by the fact that the loss of lead is caused by curtailment of travel due to slackness by wear of the gearing, and that the movements of the valve are cam-like, and are, while they last, those due to the original travel.

Long lap and long travel incur a less degree of wiredrawing than shorter lap and travel. For instance, for the Hebe and the Nile, having respectively ⅝ inch and 1 inch of lap, and 3 inches and 3¾ inches of travel, the Nile, though with the lower speed of piston, yet cutting off the earlier, shows but two-thirds of the wiredrawing in the Hebe. The Great Britain, with twice the lap and a half more travel, with a higher speed of piston and a shorter admission, shows only 3½ per cent of wiredrawing, whilst the Hebe shows from 24 per cent to 48 per cent of wiredrawing.

Pulsations of Steam in the Cylinder, and Undulations of the Indicator Diagram.

One of the most common irregularities of the indicator diagram is the initial excess of pressure with which the steam-stroke is frequently commenced, exemplified in many of the preceding figures, and strikingly developed upon the diagrams, figs. 191, 192, and 193, taken from two engines of the same class, with 24-inch stroke. Figs. 191 and 192 were both described at a speed of 25 miles per hour, or 475 feet of piston per minute. In the first case, the steam is cut off earlier than in the second, whilst the opening for preadmission at the end of the return-stroke takes place earlier. So it happens that, whereas in the latter instance there is little more than time for the entering steam to find its maximum pressure in the cylinder for the commencement of the steam-stroke, in the former instance the entrance is premature, for the steam fills the clearance while the piston has yet to complete a small part of the return-stroke. A collision takes place, a partial expulsion of the newly-admitted steam follows, and there is in consequence a great and momentary exaltation of pressure, which recovers its normal intensity before the piston describes 2 inches of the stroke. In fig. 192, as the period of preadmission is less than in fig. 191, the piston attains the end of the stroke in time to meet the entering steam

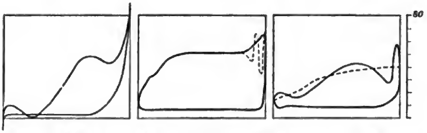


Fig. 191.—C. R., No. 124. Fig. 192.—C. R., No. 125. Fig. 193.—C. R., No. 125.
Initial Percussion.

without repelling it. The exaltation is therefore less, and the restoration of the ordinary pressure more gradual, at the same speed. At the lower speed of 15 miles per hour, or 190 feet of piston per minute, the initial disturbance in the same cylinder frequently assumes the undulatory appearance of the dot-line in fig. 192, the extra compression naturally showing itself less than at the higher speed, as there is more time allowed for adjustment to the average steam-pressure. At still lower speeds, the vertical disturbance assumes the form of a vertical line, as the adjustment is effected before the piston starts on the steam-stroke. Fig. 193 was taken with periods of compression and preadmission intermediate between those of figs. 191 and 192, and at a higher speed of 41 miles per hour, or 780 feet of piston per minute. The nipple of compression appears as a mean of those of the others. Thus, it is clear that the constant lead characteristic of the stationary-link motion, which was employed in the engines under notice, if properly adjusted for full gear, is abundantly sufficient for the timely entrance of steam, under all degrees of expansion, and at all speeds.

The originals of the indicator diagrams from the Great Britain, having the stationary-link motion, show remarkably steady steam-lines, even at the highest speeds, and under the greatest degrees of expansion; the only disturbance having been a very small initial excess of pressure of from 2 lbs. to 5 lbs. per square inch, which does not in any case extend beyond half an inch of the stroke, after which the steam-line is perfectly straight till it nears the point of cut-off. The steadiness of action is attributable to the good proportions of the cylinder and valve-gear.

As at least one class of violent vibratory movements—initial exaltations of pressure—indicated on the diagram, are pictorial transcripts of the pulsations of the entering steam, there is reason to believe that much of the subsequent oscillation observable on some diagrams is developed by the steam's elasticity. Were the undulations in all cases due to the momentum of the moving members of the indicator, they should be independent of the speed of the engine; whereas, in all cases, their periodicity coincides with the double-stroke of the piston. The M'Naught indicator employed was found to vibrate freely at the rate of 22 pulsations per second. Now, in fig. 194, described at a speed of 16½ miles per hour, 3.2 single strokes of the piston were made per second. At the same time, 3½ pulsations are indicated during the return-stroke, and they must have been delivered

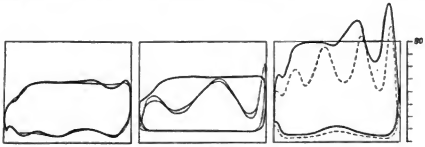


Fig. 194.—C. R., No. 124. 16½ m.p.h. Fig. 195.—C. R., No. 124. 22¼ and 28 m.p.h. Fig. 196.—C. R., No. 14. 26 m.p.h.

Pulsations.

at the rate of (3.5 × 3.2 =) 11.2 pulsations per second, which is only one-half the rate of free vibration of the indicator. It follows that the indicated undulations were, in fact, pulsations of the exhaust-steam. Fig. 195 is a compound diagram, of which the undulating figure was described at the higher speed, and it is remarkable that the undulations very closely coincide. Also, the rate and extent of pulsation is decidedly greater in this figure, during the steam-stroke, than in fig. 194 during the exhaust-stroke, and this is a difference which would not have occurred if the undulations were in all cases merely an affection of the instrument.

The influence of the want of lead in causing pulsations of steam in the cylinder is exemplified in the diagram fig. 196, from No. 14, C. R., at a speed of 26 miles per hour, or 400 feet of piston per minute, in which the lead had vanished by wear. The maximum pressure was not attained till nearly 2 inches of the stroke were described, the steam having been admitted precisely at the beginning. The first pulsation ranged through 50 lbs. of pressure per square inch, on a mean pressure of 80 lbs. This engine never worked, whilst under experimental observation, without a pulsatory action like that illustrated in fig. 196, which occasionally extended throughout the whole of the stroke, as figured in dot-lines. Now, compare this action with that of the steam in No. 13, C. R., an engine of the same class, of which the valves had their customary lead of 5/16 inch. In this engine the diagrams, fig. 197, at the same speed, 26 miles per hour, are square and solid. The pressure of steam during admission was 60 lbs. per square inch, and yet there is not a trace of agitation. Again, in the diagram, fig. 198, from the same engine, taken at a speed of 39 miles per hour, or 600 feet of piston per minute, the preliminary nipple made its appearance; but the steam-line became perfectly straight, and the exhaust was as regular. In the diagram, fig. 199, taken at 54 miles per hour, or 830 feet of piston per minute, the undulations just match in extent those of No. 14, at half the speed. These comparative facts appear to show that the want of lead is as prejudicial to the action of steam as too much of it.

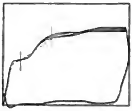


Fig. 197.—C. R., No. 13.
1st notch—26 m.p.h.

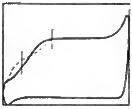


Fig. 198.—C. R., No. 13.
1st notch—39 m.p.h.

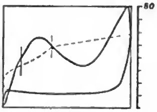
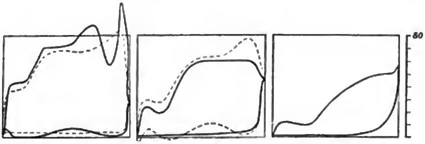


Fig. 199.—C. R., No. 13.
1st notch—54 m.p.h.

The influence of want of lead on the action of steam in the cylinder is likewise shown in the case of No. 102, C. R. At the time of the trial of this engine, her valves had entirely lost their lead in the first notch, or full-gear forward, though in mid-gear there was left 3/32 inch of lead. The sample diagrams, figs. 200, 201, 202, were taken for different degrees of expansion and for different speeds, as follows:—

			Speed of Engine.	Speed of Piston.
Fig. 200.	Dark line,	2d notch,	17½ miles per hour, ……	333 feet per minute.
	Dot line,	3d „	23¼ „ ……	442 „ „
Fig. 201.	Dark line,	4th „	21½ „ ……	409 „ „
	Dot line,	4th „	26 „ ……	494 „ „
Fig. 202. ……………		5th „	29 „ ……	551 „ „

It is apparent that at the lowest speed of all, the steam, under 2d notch, fig. 200, without any lead, made the heaviest pulsation. The 5th notch, fig. 202, at the highest speed, showed the steadiest action, attained in consequence of the presence of lead.



Figs. 200, 201, 202.—C. R., No. 102: Influence of Want of Lead. Pulsations.

The influence of water in the cylinder on the action of steam is very considerable. Water appears in the cylinders in two ways: it primes over from the boiler, and it proceeds from condensation of the steam after it is admitted into the cylinder, even, it may be, in a dry state. The direct result of priming is to increase the resistance of exhaust, and to raise the back-pressure, imposing a duty upon the steam, of pumping the water out of the cylinder at an enormous sacrifice of power. To exemplify the injurious influence of water in the cylinder, two series of diagrams were taken from the Orion, E. & G. R., immediately previous to and subsequent to blowing off of the boiler, when the water in the boiler had been unusually impure. Of these diagrams, average samples are shown, fig. 203, taken at a speed of 600 feet of piston per minute. The dark-line diagrams were taken before the boiler was blown off, and the dot-line diagrams after. The back-pressure of exhaust was, by the substitution of clean water, reduced in these instances from 9 lbs. to about 1½ lbs. per square inch. Another couple of diagrams, fig. 204, taken from the same engine on the same consecutive days, shows in a striking manner how the indicator may be affected by the presence of water and impurities in the steam.

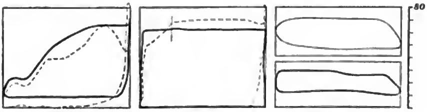


Fig. 203. E. and G. R., Orion—5th notch.
Fig. 204. Orion—1st notch.
Fig. 205. C. R., No. 73.

Influence of Water in the Cylinder.

Another example is supplied in the case of No. 73, C. R., fig. 205, which was always affected by back-pressure, caused by the exhausting of the steam into a cubical chamber situated betwixt the cylinders. The heavy line represents an average diagram taken from the cylinder, while the water in the boiler was muddy. The boiler having been blown off, and refilled with clean water, the engine turned off a better diagram, in light lining, fig. 205, in which the back-pressure, which was 12 lbs. per square inch in the first diagram, was reduced to 6 lbs.

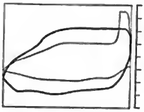


Fig. 206.
Great Britain: Influence of Water in the Cylinder.

The injurious influence of priming is rendered apparent in the case of an over-full boiler, when the Great Britain locomotive was running at a speed of 45 miles per hour. The first diagram, fig. 206, in black lining, was taken when the engine was in ordinary working order; the second diagram, in light lining, was taken when the boiler primed heavily. The diagram shows not only a reduction of the steam-line from 63 lbs. to 51 lbs., due solely to priming, but also an increase of back-pressure from an average of 15 lbs. to one of 24 lbs. per square inch.

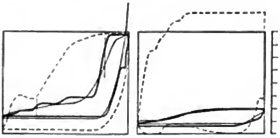


Fig. 207. Fig. 208.
C. R., No. 127: Influence of Water in the Cylinder.

In the course of some experiments with No. 127, C. R., a six-coupled wheel goods-engine with exposed ouside cylinders, 17 inches in diameter, the incidental back-pressure of exhaust was strikingly contrasted for low and high pressures, and for extreme expansive working. During a preliminary run of 20 miles, cutting off at 4 inches of a stroke of 24 inches, the diagrams in heavy and light lining, fig. 207, were taken at speeds of 17 miles and 29 miles per hour respectively, or speeds of piston of 340 feet and 580 feet per minute, after the engine had run nine miles, and had got into regular working condition, yet, here the back-pressure amounted to 14 lbs. and 12 lbs. per square inch: conspicuous instances of the influence of water in the cylinder,—arising, in this case, from internal condensation,—in aggravating back-pressure. When the engine was coupled to its train, the back-pressure, to a great extent, disappeared under larger admissions of steam and lower degrees of condensation, as exemplified by the dot-line diagram, fig. 207, taken at a speed of 23 miles per hour, or 460 feet of piston per minute, after doubling a summit, when the cylinder had been previously well heated by hard work. Again, when the engine with its tender ascended the incline alone, at 15 miles and 23 miles per hour, or 300 feet and 460 feet of piston per minute, the heavy-line and light-line diagrams, fig. 208, were taken, in which, whilst the pressure of steam admitted was 20 lbs., the back exhaust-pressures were 6 lbs. and 8 lbs. But when, on the same incline, the engine with its train ascended at a speed of 13 miles per hour, or 260 feet of piston per minute, with 95 lbs. steam admitted, the back-pressure, shown in the dot-lining diagram, fig. 208, was nothing at all,—excepting the impression of a gust of steam from the neighbouring cylinder. The contrast here is between the sluggish movements of wet steam and the nimbler movements of comparatively dry steam.

CHAPTER V.—BEHAVIOUR OF STEAM IN THE CYLINDER DURING EXPANSION.

The influence of the mass of the cylinder on the behaviour and efficiency of steam was recognized as early as 1843, when M. Combes,[1] by a remarkable forecast, formed the opinion that in steam engines a portion of the steam admitted into the cylinder was immediately condensed on the less hot metallic surfaces, which had, a second previously, been in communication with the condenser, and had had their temperature lowered in consequence; and that the condensation-water, independently of such water as might be drawn with the steam into the cylinder, was re-evaporated during the period of expansion, so that new quantities of steam were added to the steam already existing; and that the pressure fell less rapidly than in the inverse ratio of the volumes. This explanation appears to have been entirely overlooked.

Mr. Thomas Craddock,[2] in 1846, formed similar opinions, based on the fact of the constant presence of water in the cylinder, which he described as very probably occasioned by the cooling of the cylinder, in consequence of its communication with the condenser, and the condensation of part of the fresh steam admitted. He argued that the temperature of the vapour was reduced by the fall of pressure when the cylinder was opened to the condenser, below the temperature of the metal, and that consequently re-evaporation of condensation-water took place, and the cylinder was cooled down correspondingly.

Dr. Pole[3] similarly, in 1848, mentions the "accidental presence of water in the cylinder," whether in consequence of priming, or because "the temperature of the piston and the cylinder must be lower than that of the steam when first admitted, and a deposition of water will therefore take place at the commencement of the stroke." Moisture so introduced, he adds, would be re-evaporated as the steam expands.

The author,[4] unaware of what had previously been published, as matter of probability, was impressed, in the course of his investigations in 1851, on the behaviour of steam in locomotives, by the marked contrast between the expansion-curves of inside cylinders and those of outside cylinders, inasmuch as those of the former fell in pressure much more rapidly than those of the latter, and suggesting condensation in the course of the expansion, so that there appeared to be much more work done in the outside cylinder than in the inside cylinder, with steam of the same initial pressure. But, arguing that, in the inside and more or less heated cylinders, the steam must work more efficiently than in the outside and comparatively exposed cylinders, he conceived that the suggested action must be apparent rather than real, and that the comparative fulness of the diagram, and of pressure towards the end of the expansion-curve of outside cylinders, could only be derived from heat previously absorbed by the material of the cylinder from the steam when it was admitted, being in part reappropriated in resuscitating condensed steam. Having obtained this clue to the condition and behaviour of steam in cylinders, the author followed up the investigation experimentally, the results of which were published in 1851 and 1852, as already noted; and are here reproduced in abbreviated form, recalculated in accordance with the results of the more recent investigations of M. Regnault on the properties of steam.

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[1] Académie des Sciences. *Comptes rendus*, April 3d (vol. xvi. p. 649) and November 20th (vol. xvii. p. 465), 1843.

[2] *The Chemistry of the Steam Engine*, 1846.

[3] *A Treatise on the Cornish Pumping Engine*, part iii., 1848, p. 190.

[4] "Expansive Working of Steam in Locomotives," Institution of Mechanical Engineers. *Proceedings*. 1852, pp. 63, 105. Also *Railway Machinery*, 1851-55, of which the portion dealing with the investigations referred to was published in 1851.

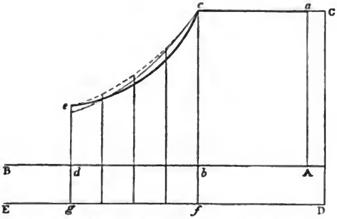


Fig. 209.—Indicator Diagram from No. 33, C. R., to Illustrate the Rate of Expansion.

In the diagram, fig. 209, the expansion of steam in the cylinder at a very low speed is exemplified. Let AB be the atmospheric line, equal to the length of the stroke of the piston, and A*b* the period of admission of steam, and the height A*a* the uniform indicated pressure, 61 lbs., maintained during admission. Then the rectangle A*c* is the area of effective pressure, due to the period of admission under the steam-line *ac*. The thick-line expansion-curve *ce* shows the progressive fall of pressure to 23 lbs. during an increase of volume, when the face of the piston has passed from *b* to the release-line *de*. The total volume of clearance between the valve and the piston amounts to 1.1 inches in length of the cylinder, and this, added to the volume described by the piston, makes up the total volume of the steam. The stroke of the piston is 20 inches, and the points of cut-off and exhaust are, by measurement of the diagram, 6.9 inches and 15 inches from the beginning of the stroke. Add to each of these the clearance, 1.1 inches, and the sums, 8 and 16.1, are the inches of stroke occupied by the initial and the final volumes of steam, at the beginning and the end of the expansion. Add the rectangular space AC, 1.1 inches wide, to represent the clearance, and draw the zero-line of pressure DE at a level 15 lbs., by scale, below the line AB, as the base-line for the expansion-curve. The total initial and final pressures of the steam during expansion, represented by *cf* and *eg*, are 76 lbs. and 38 lbs. Thus, while the total volume was increased from

8 inches to 16 inches, or doubled, the total pressure fell from 76 to 38 lbs., or to one-half. Here is a coincidence with Boyle's law of expansion at the extremities of the curve; that is, the pressure is inversely as to the volume.

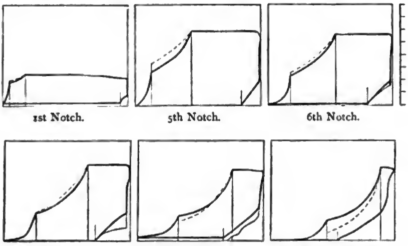
But if the period of expansion *fg* be divided into four equal parts, marked by vertical ordinates, these ordinates will measure successively 56 lbs., 47 lbs., and 41 lbs. of total pressure, for which the total volumes are 10 inches, 12 inches, and 14 inches of the stroke, respectively. By Boyle's law these pressures would be:—

For 10 inches, $76 \times \frac{8}{10} = 60.8$ lbs.
„ 12 „ $76 \times \frac{8}{12} = 50.6$ „
„ 14 „ $76 \times \frac{8}{14} = 33.4$ „

These are all in excess of the indicated pressures, and they belong to the curve *ce*, in dot lining, which properly represents the result of Boyle's law, and which touches the actual curve only at the extreme points *c* and *e*.

Again, if the steam had neither parted with nor gained heat during the expansion, it would have expanded sensibly according to the laws of simply saturated steam. The relative volume of saturated steam, of 76 lbs. absolute pressure, is 349, and this is doubled for the expanded volume, or becomes 698, which is the relative volume due to saturated steam of 36½ lbs., being 1½ lbs. less than the indicated pressure.

For the three intermediate volumes the saturated pressures are found in the same way, 60 lbs., 49½ lbs., and 42 lbs. The four pressures thus found are exhibited by the curve of saturation in light lining, fig. 209, which, for the greater part of the expansion, lies above the curve actually described, and crosses and passes below it near the extremity. It is hence to be inferred that partial condensation of steam took place during the early part of the expansion, and was succeeded by a slight re-evaporation when the temperature of the steam, which fell with the pressure, became lower than that of the cylinder.



1st Notch. 5th Notch. 6th Notch.

7th Notch. Mid Notch. Mid Notch, back Gear.

Figs. 210.—Slow Diagrams from No. 33, C. R.

The expansion-curve now discussed belongs to one of a series of indicator diagrams, figs. 210, taken consecutively from a cylinder of No. 33

passenger locomotive on the Caledonian Railway, with different periods of admission and expansion, as regulated by the notches of the valve-gear. The ratios of the initial to the final volumes expanded, clearance included, are from 1 to 1.14, to 1 to 3½, and the final pressures by Boyle's law vary from 2 lbs. in excess to 10 lbs. in deficit, compared with the indicated pressure. The curves of expansion due to Boyle are shown in dotting, and, taking them as approximately indicative of the lines of expansion, which would have been described had there not been any of the successive condensation and re-evaporation already referred to, it appears that, since the actual curve sinks below the dotted curve in the first stages of expansion, and rises above it in the later stages, if sufficiently prolonged, the steam was condensed, and then re-evaporated, so that occasionally there is more sensible steam at the end than at the beginning. The mass of the cylinder is, in short, an equalizer tending to reduce inequalities of temperature and pressure.

These diagrams, figs. 210, were taken after the engine had been at rest for some hours, when the cylinder had cooled considerably. The loss by condensation of steam admitted into comparatively cool cylinders is strikingly exemplified by the indicator diagram, fig. 211, taken from No. 125, C. R., a goods engine with outside cylinders, immediately at starting, after two hours of inactivity. The diagram presents a falling pressure during admission, the result of condensation; whereas, had the cylinder been quite hot, the curve would have followed the dotted line. Again, the diagram, fig. 212, from the same cylinder, cutting off at 3½ inches of the stroke, shows a large resuscitation of steam during the later part of the expansion.

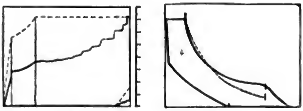
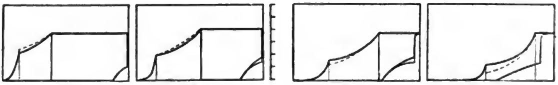


Fig. 211. Fig. 212.

Indicator Diagrams from No. 125, C. R., to show Condensation of Steam.

The slow diagrams, figs. 213, were taken from No. 13, C. R., immediately after having made an express trip, while yet the cylinders were hot. They are sharp and square, and exhibit nothing of the indecision noticeable in those taken from No. 33, figs. 210. The dotted expansion lines show the curves according to Boyle's law, by which the pressures are inversely as the volumes; and in the hotter cylinder of No. 13 an expansion into one and a half times the initial volume brings up the pressure to an equality with that which is calculated by Boyle's law, as illustrated by the dot curves; though in No. 33, having the colder cylinder, an expansion into double the volume was required for the same result.



1st Notch. 2d Notch. 3d Notch. Mid Notch.

Figs. 213.—Slow Indicator Diagrams from No. 13, C. R.

Of the Expansion of Steam in the Cylinder during Regular Work.

The types of locomotives, the indicator diagrams from which are now to be discussed, are shown in cross section through the cylinders and the smoke-box, in the figures 188, 189, 190, page 457, in connection with which the principal dimensions are given.

The table No. 99, founded on the indicator diagrams taken from the Great Britain, shown in Plate I., exhibits the manner in which they have been analysed in relation to the expansion-lines. The points of cut-off and exhaust, including 1.80 lineal inches for clearance, with the initial and final volumes for expansion, are given in the subjoined tablet. The volumes are calculated as the product of the area of the piston, 1.767 square feet, by the periods of total admission, and the lineal measures of the total expanded volumes, in feet.

GREAT BRITAIN.	Cut-off plus Clearance.	Exhaust plus Clearance.	Initial Volume.	Final Volume.	Actual Ratios of Expansion.
Notch.	inches or feet.	inches or feet.	cubic feet.	cubic feet.	ratios.
No. 1	17.8 or 1.483	22.8[1] or 1.900	2.620	3.357	1 to 1.30
„ 3	13.8[2] „ 1.150	20.8[2] „ 1.733	2.032	3.062	1 „ 1.50
„ 5	8.8 „ 0.733	18.8[3] „ 1.567	1.295	2.769	1 „ 2.14

[1] The period of exhaust is taken at 21 inches, for a whole number, plus 1.8 inches.

[2] The points of cut-off and exhaust are taken at 12 inches and 19 inches, plus 1.8 inches each.

[3] The point of exhaust is taken at 17 inches.

The observed speeds of the engine, and the calculated speeds of the piston, are stated in columns 2 and 3 of table No. 99. The sensible initial and final pressures of the steam, above the atmosphere, measured from the diagrams, are stated in columns 4 and 5. The initial and final weights of steam, assumed meantime to be saturated, columns 6 and 7, are calculated by multiplying the total volumes of the steams respectively, including clearance, by their densities or weights per cubic foot. Lastly, the differences of the initial and final weights of sensible steam, with their values as percentages of the initial weights of steam, are given in columns 8 and 9, the percentage in excess being marked + (plus), and those in deficiency marked − (minus).

It appears from the table that, in nearly all instances, there is less sensible steam, taken as saturated, at the end of the expansion than at the beginning; also, that the influence of speed on the difference of the weight of steam is not appreciable except under the first and fifth notches, when at the two lowest speeds the final weight of steam is nearly the same as, or greater than, the initial weight. In these instances, it is intimated that preliminary condensation in a comparatively cool cylinder, and subsequent re-evaporation, took place. On the contrary, the reduction of weight of steam during expansion, when the cylinder is well warmed up, expressed in parts of the initial weight, increases as the ratio of expansion is increased, the average reduction amounting to 5.05 per cent, 7.55 per cent, and 9.20 per cent, for the first notch, third notch, and fifth notch, respectively.

Table No. 99.—GREAT BRITAIN LOCOMOTIVE—EXPANSION AND WEIGHT OF STEAM IN THE CYLINDER, FROM INDICATOR DIAGRAMS. 1850.

Cylinder 18 inches × 24 inches; total clearance 1.8 inches; wheel 8 feet. Steam assumed to be saturated, for calculation.

Number of Diagram.	Speed of		Net Pressure per Square Inch during Expansion.		Weight of Sensible Steam, as Saturated.			
	Engine.	Piston.	Initial.	Final.	Initial.	Final.	Difference.	
1	2	3	4	5	6	7	8	9
No.	miles per hour.	feet per minute.	lbs.	lbs.	lb.	lb.	lb.	per cent.
FIRST NOTCH: Cut-off at 67 per cent; exhaust at 89 per cent; actual ratio of expansion, 1 to 1.30.								
25	15	210	70	50	0.5188	0.5163	0.0025	– 0.48
26	17	238	88	65	0.6218	0.6275	0.0057	+ 0.92
27	21	294	94	65	0.6549	0.6275	0.0274	– 4.18
28	24	336	84	57	0.5988	0.5681	0.0307	– 5.13
29	27	378	74	49	0.5416	0.5090	0.0326	– 6.02
30	31	434	86	55	0.6103	0.5533	0.0570	– 9.34
31	31	434	80	56	0.5760	0.5607	0.0153	– 2.66
32	49	686	52	34	0.4148	0.3958	0.0190	– 4.58
33	54	756	86	60	0.6103	0.5905	0.0198	– 3.24
Averages Nos. 27–33	34	474	79.4	53.7	0.5724	0.5436	0.0288	– 5.05
THIRD NOTCH: Cut-off at 50 per cent; exhaust at 82 per cent; actual ratio of expansion, 1.50.								
34	17	238	87	48	0.4777	0.4570	0.0207	– 4.33
35	18	252	70	37	0.4024	0.3953	0.0071	– 1.76
36	21	294	90	48	0.4905	0.4572	0.0333	– 6.79
37	26	364	72	38	0.4113	0.3886	0.0227	– 5.52
38	31	434	75	38	0.4245	0.3886	0.0359	– 8.46
39	32	448	79	38	0.4422	0.3886	0.0536	– 11.58
40	40	560	65	33	0.3798	0.3540	0.0258	– 6.79
41	51	714	55	28	0.3349	0.3191	0.0158	– 4.72
42	55	770	72	37	0.4113	0.3816	0.0297	– 7.22
Averages Nos. 36–42	37	512	72.6	37.1	0.4133	0.3821	0.0312	– 7.55
FIFTH NOTCH: Cut-off at 29 per cent; exhaust at 72.5 per cent; actual ratio of expansion, 1 to 2.14.								
43	17	238	89	33	0.3099	0.3201	0.0102	+ 3.29
44	18	252	70	24	0.2565	0.2636	0.0071	+ 2.77
45	21	294	93	33	0.3208	0.3201	0.0007	– 0.22
46	28	392	74	21	0.2677	0.2439	0.0238	– 8.89
47	31	434	80	22	0.2847	0.2506	0.0341	– 11.98
48	36	504	63	17	0.2365	0.2185	0.0180	– 7.61
49	50	700	55	15	0.2135	0.2057	0.0078	– 3.65
50	56	784	65	16	0.2421	0.2121	0.0300	– 12.39
Averages Nos. 46–50	40	563	67.4	18.2	0.2489	0.2260	0.0229	– 9.20

In the abstract table, No. 100, which follows, therefore, the speeds are omitted, as unnecessary for making comparisons, but the points of cut-oft and exhaust, and the actual ratios of expansion, are given. The contents of this table are based on the average results of detailed tables compiled for each of the engines named, similar to table No. 99 for the Great Britain, derived from the evidence of several indicator diagrams in each case, taken from the engines whilst on regular duty. Inside cylinders and outside cylinders are classed separately. The results from the other inside cylinders are corroborative of those from the Great Britain; and it is shown that, for an average cut-off of 52.4 per cent, or little more than half-stroke, the average reduction of weight of steam, taken as saturated during expansion, is 6.27 per cent.

Table No. 100.—EXPANSION AND WEIGHT OF STEAM IN THE CYLINDERS OF LOCOMOTIVES, INSIDE AND OUTSIDE, FROM INDICATOR DIAGRAMS. 1850.

INSIDE CYLINDERS.

NAME OF ENGINE.	No. of Notch.	Cut-off.	Ex-haust.	Actual Ratio of Expansion.	Net Pressure per Square Inch during Expansion.		Weight of Sensible Steam, taken as Saturated.			
					Initial.	Final.	Initial.	Final.	Differences.	
1	2	3	4	5	6	7	8	9	10	11
	No.	per cent.	per cent.	ratio.	lbs.	lbs.	lb.	lb.	lb.	per cent.
Great Britain...	1	67.0	89.0	1.30	79.4	53.7	0.5724	0.5436	0.0288	− 5.05
Do. ...	3	50.0	72.0	1.50	72.6	37.1	0.4133	0.3821	0.0312	− 7.55
Do. ...	5	29.0	72.5	2.14	67.4	18.2	0.2489	0.2260	0.0229	− 9.20
Nile...............	Gab.	67.5	89.5	1.30	36.4	22.1	0.1921	0.1833	0.0088	− 4.57
Orion	5	56.7	89.0	1.50	46.5	21.5	0.1924	0.1806	0.0118	− 6.13
Hebe..............	5	44.4	77.0	1.64	46.0	19.0	0.1496	0.1417	0.0079	− 5.28
Averages.........	...	52.4	...	1.56	...	...	...	...	...	− 6.27
OUTSIDE CYLINDERS.										
C. R., No. 13....	1	62.6	86.5	1.35	52.0	34.0	0.2269	0.2268	0.0001	− 0.04
Do. ...	2	49.0	81.5	1.60	45.0	24.0	0.1644	0.1735	0.0091	+ 5.54
C. R., No. 33....	5	54.5	86.0	1.52	56.0	27.0	0.2049	0.1909	0.0140	− 6.83
Do. ...	7	34.5	75.0	2.01	46.0	16.5	0.1184	0.1281	0.0097	+ 8.19
C. R., No. 41....	1	70.4	91.0	1.26	36.0	24.0	0.1940	0.1908	0.0032	− 1.65
Do. ...	2	64.0	86.0	1.31	26.0	16.0	0.1446	0.1457	0.0011	+ 0.77
C. R., No. 42....	3	53.4	84.5	1.52	55.0	30.0	0.2038	0.2043	0.0005	+ 0.24
Do. ...	4	43.4	78.5	1.70	49.0	22.0	0.1566	0.1582	0.0016	+ 1.02
Do. ...	5	15.4	68.5	3.33	68.0	22.0	0.0890	0.1397	0.0507	+ 56.94
C. R., No. 51....	3	45.6	82.0	1.72	59.0	29.0	0.1838	0.1927	0.0089	+ 4.84
Do. ...	4	28.6	70.5	2.20	50.0	20.0	0.1117	0.1360	0.0243	+ 21.75
C. R., No. 125...	2	72.0	92.5	1.27	33.0	20.0	0.2857	0.2659	0.0168	− 5.88
Do. ...	3	55.4	89.0	1.54	48.0	26.5	0.2924	0.3071	0.0147	+ 5.03
Do. ...	4	36.0	85.5	2.16	37.0	13.5	0.1670	0.2057	0.0387	+ 23.17
C. R., No. 127...	5	19.0	61.0	2.84	57.0	20.0	0.1267	0.1826	0.0559	+ 44.10
Averages.........	...	47.0	...	1.82	...	...	...	...	...	+ 10.08

With respect to the differences of the weights of sensible steam at the points of cut-off and exhaust, in the inside cylinders, calculated on the supposition that the steam was saturated during the whole period of expansion, the fact that the final weight thus calculated is less than the initial weight is evidence that the steam was, in fact, not saturated but superheated, and that it behaved as a permanent gas during expansion, adiabatically.

The action of the steam in the outside cylinders is broadly distinguished from that of the steam in the inside cylinders. It is shown that, for an average admission of 47 per cent, and an average expansion-ratio of 1.82, there is a positive excess of 10.08 per cent of steam at the end of the expansion above the initial weight—contrasting with the negative difference for the inside cylinders. But, in four instances of low ratios of expansion, with long periods of admission—of Nos. 13, 33, 41, and 125—the final weight of steam is less than the initial weight. These differences are to be interpreted in exactly the contrary sense to those of the inside cylinders, for they signify that the steam was partly condensed during expansion correspondingly to the condensation already shown to have taken place in outside cylinders in the earlier parts of the expansion period, and which was succeeded by re-evaporation in the later part. It is clearly indicated by the several groups of results for each engine that, whilst the initial weight of sensible steam is reduced as the period of admission is reduced, the final excess of weight is absolutely increased. For instance, in C. R., No. 42—

Cutting off at..............................	53.4	43.4	15.4 per cent,
The initial weight of steam cut off is..	0.2038	0.1566	0.0890 lbs.,
The final excess weight of steam is...	0.0005	0.0016	0.0507 „
Being, in parts of the initial steam.....	+0.24	+1.02	+56.94 per cent.

The increase of the weight of re-evaporated steam, previously condensed, at the end of the expansion, in excess of the weight at the beginning of it, as the steam is cut off earlier and expanded more, in outside cylinders, is graphically represented in the diagram, fig. 214, in which the base-line AB represents the length of the stroke, and is divided into ten parts of 10 per cent each. The points of cut-off in the third column of table 100, for outside cylinders, are set off on the base-line from B, and from the points so found ordinates are drawn to the base-line, on which the percentages of excess-weight are set off according to the vertical scale, and indicated by crosses. The medium curve CDA′, traced through these crosses, represents the average indicated excess-weight of steam at the end of the expansion according to the period of admission. The negative excesses, or positive deficiencies, are plotted below the base-line. It is shown that in cutting off at 10 per cent of the stroke there is 80 per cent excess final weight, which diminishes as the period of admission increases, until, in cutting off at 53 per cent of the stroke, the excess vanishes. For longer periods of admission there are deficiencies of weight, as represented by the dotted portion of the curve below the base-line. This curve will bear interpretation. The negative section of the curve indicates positive condensation as well as the positive section, and the curve of actual condensation must be above the base-line for the whole length. The negative quantities marked by crosses may be plotted at equal distances above the base-line, as represented by small squares. There are plotted in addition, marked by circles, five actual results of trials with Nos. 42 and 48, C.R., of which an account will be given. The curve has therefore been rectified by the insertion of the solid line in substitution for the lower part of the curve, forming the curve CA′, and interpreting and embodying more directly the experimental facts of condensation.

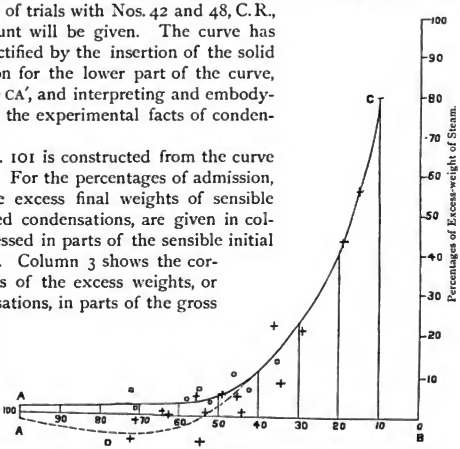


Fig. 214.—Diagram to show Initial Condensation of Steam in Unprotected Cylinders, worked expansively.

The table No. 101 is constructed from the curve as finally settled. For the percentages of admission, in column 1, the excess final weights of sensible steam, or indicated condensations, are given in column 2, and expressed in parts of the sensible initial weights of steam. Column 3 shows the corresponding values of the excess weights, or indicated condensations, in parts of the gross weights of steam admitted into the cylinder, which are taken as expressed by the final weights, comprising the initial sensible steam plus the positive excess-weight of steam or indicated condensation. This column is intended to show the indicated proportion condensed of the whole steam admitted. Column 4 shows the corresponding actual ratios of expansion reckoned to the point of exhaust. These were ascertained by plotting the volumes of expansion expressed by ordinates to a base-line representing the length of the stroke, and tracing a curve through them from which the values in column 4 were measured off. Column 5 shows the actual ratios of expansion reckoned to the end of the stroke. By columns 6 and 7 it is designed to show the relative absolute weights of steam condensed for the different periods of admission. For this object, the total volumes of steam cut off are expressed in column 6 by the period of admission plus the clearance, which is here taken as 5 per cent of the stroke; and the relative absolute weights of steam condensed are expressed in column 7, in which the absolute weight of steam condensed for 10 per cent admission, or 15 per cent total admission, is taken as 1. The relative absolute values for this and the other admissions, being directly as the total periods of admission, column 6, and as the percentages of condensation, column 3, they are expressed by the products of these quantities. For instance, for 10 per cent admission, or 15 per cent total admission, the steam condensed is 44 per cent of the whole steam admitted, and it is absolutely as (15 × 44 =) 660. For 15 per cent admission, the steam condensed is 36 per cent, and it is absolutely as (20 × 36 =) 720. If, then, the value 660 be taken as 1, the value 720 is (720 ÷ 660 =) 1.09. These and the corresponding values for the relative absolute weights of steam condensed, for the other periods of admission, are as ranged in column 7.

Table No. 101.—Outside Cylinder Locomotives, C. R.

Indicated Proportion of Steam Condensed in the Cylinder for various Periods of Admission. 1850.

Minimum values, being values as indicated.

Periods of Admission, in parts of the Stroke. 1	Excess Final Weight of Steam, or Indicated Proportion of Steam condensed, in parts of Initial Sensible Weight admitted. 2	Indicated Proportion of Steam condensed, in parts of Gross Indicated Weight of Steam admitted. 3	Actual Ratio of Expansion to Point of Exhaust. 4	Actual Ratio of Expansion to End of Stroke. 5	Period of Admission, plus 5 per cent Clearance. 6	Relative Absolute Weights of Steam Condensed; that for 10 per cent admission being taken = 1. 7
per cent.	per cent.	per cent.	ratio.	ratio.	per cent.	
10	80.0	44.0	4.00	7.00	15	1
15	57.0	36.0	3.40	5.25	20	1.09
20	41.0	29.0	2.85	4.20	25	1.10
25	31.0	23.6	2.50	3.50	30	1.07
30	23.0	18.7	2.20	3.00	35	.99
35	17.5	15.0	2.00	2.62	40	.91
40	11.0	10.0	1.83	2.33	45	.68
45	7.5	7.0	1.70	2.10	50	.53
50	4.5	4.3	1.60	1.91	55	.36
55	3.5	3.4	1.48	1.75	60	.31
60	3.0	2.9	1.40	1.62	65	.29
70	2.75	2.7	1.25	1.40	75	.30
80	2.5	2.4	1.15	1.24	85	.31
90	2.25	2.2	1.075	1.11	95	.32
100	2.0	2.0	1.00	1.00	105	.32

With reference to this table and the data upon which it is based, the proportions of steam condensed, as indicated, are to be taken as minimum values deduced from the indicated preponderance in weight of steam at the end of the expansion over the weight of steam as cut off. The actual values for the higher degrees of expansion are certainly greater than the indicated values, for the reasons already given. The relative absolute weights of steam condensed, as in column 7, exhibit, in a striking manner, the formidable degrees of condensation that prevail for the shorter admissions. It may be taken that for all periods of admission shorter than 35 per cent the weights of steam condensed are absolutely the same, although the total weights admitted may be greatly reduced.

The author's conclusions from the foregoing investigations of the behaviour of steam in the cylinder during expansion, were summarized as follows, in 1851, in *Railway Machinery*, from which quotations are made:—

"In general, it is to be concluded that, first, when the cylinder is thoroughly immersed in the hot-bath of the smoke-box, the temperature of which is commonly much higher than that of the steam, the quantity of water existing as steam during expansion is virtually constant. Secondly, when the cylinder is placed nearly or entirely beyond the influence of the heat of the smoke-box, or is protected only in the usual manner by felting and plating, the quantity of water or steam varies very considerably during expansion. It suffers a rapid and transient diminution during the first stages of expansion, and remounts to an excess over the initial quantity, which increases uniformly as the ratio of expansion is prolonged; till, for a final volume of about 3½ times the initial volume, when the steam is cut off at 15 per cent, the excess amounts to 57 per cent of the weight of sensible steam cut off, or, otherwise, to 36 per cent of the gross quantity of steam admitted."

"The foregoing results are directly contrary to what might have been anticipated, as, at first sight, they appear to show that the less protected the cylinders, the more work is done with a given initial quantity of steam. In the inside cylinder, so far from any apparent evaporation or accession to the total weight of the steam during expansion, the quantity is at least not more than constant, and is, in fact, slightly reduced during expansion. The outside cylinders, on the contrary, show, by the great excess of steam at the end of expansion, very significant amounts of factitious evaporation. In this case, as in that of the low-speed diagrams, the difference is referable to a primary condensation of the steam during admission, by which water is formed, whilst the temperature of the cylinder is raised. After suppression, and when the steam temperature falls by expansion below the newly-acquired temperature of the cylinders, the hot water flashes into steam in virtue of its own heat and that of the cylinder, according to the law of the maximum density and pressure for the temperature; and what appears, at first sight, to have been positively one advantage of an exposed cylinder, in the auxiliary evaporation during the later stages of expansion, is nothing more than a partial resuscitation of the precipitated steam, as a compromise for lost initial action. The greater the proportion of expansion, the greater is the final excess of steam, as the extreme temperatures become more widely different; and, moreover, for the higher degrees of expansion, smaller absolute volumes of steam are admitted, for which there is always the same cooling superficies of cylinder; and this is relatively greater, of course, as the period of admission is reduced. In the inclosed inside cylinder, on the contrary, bathed in hot air or enveloped in cinders, actually hotter than the steam that passes through them, the initial pressure of the steam as it enters the cylinder is maintained in its integrity, as, even for the greater expansions, there appear no symptoms of a resurrection of steam. The evidence goes rather to show that the steam is superheated during its passage through the steam-pipes previous to admission.

"An elegant confirmation of these conclusions on the condensation of steam insufficiently protected, is derivable even from the inside cylinders of locomotives themselves. It is known by observation that, whatever be the temperature in the smoke-box during the action of the blast, it falls very rapidly to about 200° F. when the steam is shut off. On starting from a station, therefore, the atmosphere of the smoke-box requires raising in temperature, which involves a little time for its accomplishment. During this interval, the cylinder is colder than the steam, but is becoming gradually hotter—a process which may be plainly traced over a succession of indicator diagrams, taken at short intervals after starting. For example, on the occasion of an express trip from Edinburgh to Glasgow with the Nile, E. and G. R., of which the average results are given in table No. 100, the train, starting from Linlithgow station, passed the mile-posts at the intervals and at the speeds given in the following table, No. 102, in the last column of which are contained the successive differences of weights of steam at the beginning and the end of the expansion. The steam had been shut off for a period of 3 minutes 40 seconds before the engine started; and though this period of inactivity seems very short, yet the engine was, as usual, pumping air into and out of the cylinders after the steam was shut off, and cooled down before the engine arrived at the station.

Table No. 102.—NILE, E. and G. R.: GRADUAL REHEATING OF THE CYLINDER AFTER LEAVING LINLITHGOW. 1850.

Mile-posts.	Time of Passing Mile-posts.			Speed of Passing Mile-posts.	Difference of Weights of Initial and Final Steams.
	h.	m.	s.	miles pr. hour.	per cent.
Linlithgow depart—16½ mile-post............	11	8	40	...	...
„ „ 17 „		11	10	15.0	+0.13
„ „ 17½ „		12	35	22.5	−1.18
„ „ 18 „		13	50	24.0	+1.70
„ „ 19 „		16	12	27.0	−1.33
„ „ 20 „		18	17	30.0	−2.51
„ „ 21 „		20	12	32.0	−4.46
„ „ 22 „		22	0	34.0	−4.76
„ „ 23 „		23	40	34.0	−3.94
„ „ 23½ „		24	30	34.0	−5.12

"This table shows that the train had run nearly two miles before the positive differences of weight, indicating internal condensation and re-evaporation, disappeared; and only when the engine reached the twenty-first mile-post, after a run of 5 miles, during an interval of 11½ minutes, did it appear to have got into regular order, when the negative differences of weight in the last column, which indicated a superheated cylinder, had ceased to augment, and became practically constant, by a process due to the gradual rise of the temperature in the smoke-box about the cylinders."

"The loss by condensation proved to have been incurred in the outside cylinder, great as it has been proved to be, for the shorter periods of admission is, after all, only the obvious loss discoverable in the indicator diagram, which is certainly less than the actual loss; for it cannot be assumed that the final weight of steam, even for long periods of expansion and short admissions, represents the total quantity of steam that enters the cylinder. There must necessarily be an extra constant percentage of loss by condensation, which never appears in the diagram, and is dissipated through the sides of the cylinder, setting out, as we do, with the proposition, now established, that the outside or imperfectly protected cylinder is partially a condenser. The heat so dissipated leaves in the cylinder a constant quantity of water, which is exhausted with the steam, and only makes itself felt in the diagram by an exaggeration of the back-pressure.

"Though it has been found, therefore, that an expansion into a volume of 1.50—the steam being cut off at 53 per cent of the stroke—indicates equal weights of steam, initial and final, it does not prove the absence of condensation, any more than an expansion of 1.40, steam being cut off at 60 per cent, indicating a final weight less by 2½ per cent than the initial weight, proves, under the given conditions, a case of initial superheating, such as was fairly inferred for inside cylinders. The equality of the initial and final weights, in the first case, simply shows that the initial condensation continues until the expansion is partly accomplished, and is succeeded by re-evaporation during the remainder of that period. For the shorter expansion 1.40, and, of course, for still smaller expansions, with larger admissions, the quantity of water re-evaporated being insufficient to restore equality of initial and final weights, an inferior final weight of steam is the result. And although in these cases the condensation, less obvious on the expansion-curve, is not the less a reality, it is probable that the constant condensation is less in amount than for greater expansions. . . . Seeing, then, that the indicated expansion-curve leaves us in a state of uncertainty as to the whole loss of effect by condensation, though we know that what we do discover by an examination of the curve is certainly less than the actual loss, we can at present only regard the ordinary loss by initial condensation in outside cylinders, working with 35 per cent of admission—a common practice on ordinary railways—as at least 15 per cent of the steam delivered into the cylinder. It is certainly more; and the amount of the real loss is matter for further consideration. To cause initial condensation, it is not even necessary to assume that there is any external dissipation of heat, for, however well a cylinder may be protected from escape of heat externally, it must operate as an equalizer of heat by, in the first place, condensing a part of the steam as it enters, and, secondly, partially restoring the heat only when the pressure has fallen. It is therefore necessary for the perfect employment of steam in the cylinder to maintain the temperature of the latter at least equal to the initial temperature of the steam by independent means applied without the cylinder, in place of ruinously drawing upon the constituent heat of the steam within."

"These results sufficiently explain how it happens that expansive working in locomotives, especially in outside-cylinder engines, is in practice carried out to such a limited extent. We have rarely found (1850) on the Caledonian Railway—a line stocked with outside-cylinder locomotives—that a cut-off materially less than 30 per cent of the stroke is voluntarily adopted by engine-drivers. In their own words, they 'lose as much as they gain' if they endeavour to work with a suppression much less than 30 per cent; and yet it is not easy to conceive a line better adapted than the Caledonian, composed of long and steep double-gradients, for the practice of highly expansive working. The balance of loss and gain above noted is abundantly explained by the extra condensation which attends the earlier cut-off, and is not at all referable to the popular notion that there is something wrong in the nature of the link-motion. We may refer to a case in point—the case of the locomotives on that railway, Nos. 25 and 51, taking the same train. As much water was consumed from the tender of No. 51—of which the driver was one of the most economical consumers of coke on the line, and which was worked with variable expansion within the limit already stated—as was withdrawn from the tender of No. 25, an engine worked mostly by the regulator; whereas, steam of a higher pressure, if worked expansively under proper conditions, should have consumed a less quantity of water than wire-drawn steam in the performance of the same duty. There is no doubt that much of the economy of fuel effected with No. 51 was due to the judicious system of stoking practised by the driver, and to his unceasing care in preventing waste of steam at the safety-valve—matters to which little attention was paid with No. 25.

"Nothing is more common than a rush of water when the outside cylinder is tapped for the indicator, whilst the steam is worked with a considerable degree of expansion. With the goods engine, No. 125, C. R., of which the cylinders (fig. 190, p. 457) are still more exposed than those of No. 51 (fig. 189), the cylinders, during one experiment, were never free from a stream of water through the indicator, even for long-continued runs; and during temporary stoppages large accumulations of water were usually formed, which apparently could never be entirely dissipated, even through the open cylinder-cocks."[1]

Independent evidence of the condensation of steam in exposed cylinders is deduced from a comparison of the indicated consumption of steam as cut-off, with the measured consumption of water from the tender.[2] For the purpose of testing such comparison, the following, amongst other special trials, were made by the author, August 29th, 1850, on the actual consumption of water by the outside-cylinder express engine, No. 42, on the Caledonian Railway, for a trip of 105 miles from Glasgow to Carlisle, with a train averaging six and a half carriages, the time of the trip being three hours twenty-two minutes, including five stoppages. Indicator diagrams were taken from one of the cylinders at intervals of

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[1] This and the immediately preceding paragraphs are quoted from the earlier portion of *Railway Machinery*, pp. 82–84, published in 1851.

[2] Derived from *Railway Machinery*, pp. 145 to 148.

1 mile or 2 miles, the notch of the expansion-gear was noted for each diagram, and also the points of the line where each change of expansion was made. The periods of admission for the front stroke are given in the following tablet, column 2. The mean total admissions for the front and back strokes, including a clearance of 1.1 inches, and the mean total periods of compression, are given in columns 3 and 4. The volume of steam admitted per mile run under each notch, and likewise the volume shut in and compressed, columns 5 and 6, have been calculated in terms of the area of the piston, the total admissions and compressions in columns 3 and 4, and the number of turns of the driving-wheel in traversing 1 mile of way.

C. R., No. 42, PASSENGER LOCOMOTIVE.

Notch. 1	Admission, Front Stroke. 2	Mean Total Admission, including Clearance. 3	Mean Total Compression, including Clearance. 4	Volume admitted per Mile. 5	Volume Compressed per Mile. 6
	inches.	inches.	inches.	cubic feet.	cubic feet.
1	14.5	15.1	...	...	...
2	13.0	13.6	4.1	1,558	469
3	11.0	11.7	4.5	1,340	515
4	9.0	9.35	5.3	1,128	607
5	3.4	4.5	8.1	515	916

The results of the working of the steam are given in abstract in table No. 103, page 480, in four sections, for each of which individually the working of the engine has been analysed.

To calculate the quantity of water as sensible steam, taken as saturated, passed into the cylinder, the volume admitted per mile is divided by the relative volume of the steam according to M. Regnault's data, at the pressure when cut off, and the quotient is the equivalent volume as water admitted. Similarly, the quantity of water as steam detained by compression is found by dividing the volume of steam inclosed by its relative volume. The net volume of water as steam passed through the engine for one end of one cylinder, is the difference of the volumes admitted and compressed. For instance, in the first section, for a length of ½ mile, steam of 47 lbs. pressure is cut off under the second notch, of which the relative volume is 424. The mean volume of total admission per mile-run is, by the above tablet, 1558 cubic feet; and 1558÷424=3.68 cubic feet of water as steam cut off per mile. For ½ mile it is (3.68 × ½ =) 1.84 cubic feet of water as steam. The steam compressed in the return-stroke is shut in at a pressure of 4 lbs. per square inch, for which the relative volume is 1290, and the volume shut in per mile is by the tablet, 469 cubic feet; then 469÷1290=0.36 cubic foot, and, for ½ mile, 0.18 cubic foot. Deducting the volume thus detained from the volume cut off, the difference is (1.84−0.18=) 1.66 cubic feet of water as sensible steam consumed. Treating in like manner the three other entries in the first section of table No. 103, the results are, in brief, as shown in the table on following page:—

Table No. 103.—WORKING OF THE STEAM IN THE PASSENGER LOCOMOTIVE, NO. 42, C. R., WITH AN EXPRESS TRAIN, AUGUST 29, 1850, FROM GLASGOW TO CARLISLE.

Stations, Intermediate Distances, and Times of Steam up. 1	Notch under which the Engine was worked. 2	Miles run under each Notch with Steam on. 3	Average Indicated Pressure per Square Inch.	
			At Cut-off. 4	At Compression. 5
	No.	miles.	lbs.	lbs.
(1.) Glasgow to Motherwell, 16 miles, steam on 30 minutes.	2	½	47	4
	4	5½	36	5
	4	5½	50	5
	4	3	72	5
		14½		
(2.) Motherwell to Carstairs, 15½ miles, steam on 29½ minutes.	3	10	69	4
	3	2¼	38	3
	4	2	50	5
		14¼		
(3.) Carstairs to Beattock, 34 miles, steam on 38¼ minutes.	3	9½	50	4
	4	11	50	5
	4	3	58	5
		23½		
(4.) Beattock to Carlisle, 39½ miles, steam on 56¼ minutes.	2	½	60	4
	3	3	60	4
	3	1¼	30	3
	4	12¾	38	5
	4	2	50	5
	4	1¾	33	4
	4	7½	52	5
	5	3½	30	3
	5	6½	56	3
		38¾		

NO. 42, C. R., SECTION (1).

	Miles Run.	Cut-off Pressure.	Water as Steam Cut off.	Shut-in Pressure.	Water as Steam Compressed.	Difference, Water as Steam Consumed.
		lbs.	cubic feet.	lbs.	cubic feet.	cubic feet.
2d Notch	½	47	1.84	4	0.18	1.66
4th ,,	5½	36	12.43	5	6.92	31.78
,, ,,	5½	50	15.29			
,, ,,	3	72	10.98			
			38.70			
Water as steam consumed.....................						33.44

The net indicated quantities of water as steam consumed for the three other sections of the trip are calculated likewise. The following table, No. 104, contains these and other data for comparison. In columns 7 and 8 are given the indicated quantities of steam condensed, being the differences between the quantities of water consumed from the tender, column 4, and the indicated quantities of water as steam cut off, less the steam retained by compression, column 6. In columns 9 and 10 the indicated quantities of steam condensed according to the diagram, fig. 214, page 473, and the corresponding table, No. 101, page 474, are given for comparison with the quantities, columns 7 and 8, as deduced from direct measurement from the tender. The correspondence is very remarkable, and shows as near an approximation to equality as could in reason be expected under the circumstances. The increase of indicated condensation, column 8, as the period of admission, column 2, is reduced, is apparent. Let the means of sections (1) and (3), which have nearly equal periods of admission, be taken; then, with sections (2) and (4), the following series is obtained:—

Table No. 104.—Water Consumed and Steam Condensed in the Working of Passenger Engine, C. R., No. 42, with an Express Train. 1850.

Number of Section.	Average Mean Periods of Admission.	Time of Actual Evaporation.	Water Consumed from the Tender.		Net Indicated Water as Steam consumed at Cut-off, less Compression.	Indicated Quantities of Steam Condensed, Difference of columns 4 and 6.		Indicated Quantities of Steam Condensed, according to Table 101, page 474, column 3.	
			Total.	Per Hour.					
	2	3	4	5	6	7	8	9	10
No.	per cent.	minutes.	cub. feet.	cub. feet.	cubic feet.	cub. feet.	per cent.	cub. feet.	per cent.
1	42	30	35.82	71.6	33.44	2.38	6.6	3.53	10.0
2	52	29½	48.85	99.4	47.16	1.69	3.5	2.20	4.5
3	46	38¼	67.74	106.2	60.69	7.05	10.4	5.08	7.5
4	36	56¼	79.50	84.8	69.04	10.46	13.2	12.22	16.0
	42	154	231.91	93.6	210.33	21.58	9.3	23.20	10.0

Section.	Cut-off.	Water Consumed per Hour.	Steam Condensed.	Do. per Table No. 101.
	per cent.	cubic feet.	per cent.	per cent.
(2)	52	99.4	3.5	4.5
(1) & (3)	44	88.9	8.5	8.75
(4)	36	84.8	13.2	16.0

The marked variation of condensation with the cut-off inversely, and the close correspondence of the condensations, as calculated in the two modes, from the consumption of water from the tender, and from the internal evidence of indicator diagrams, table No. 104, are proofs of the fact of the condensation of steam in the cylinder, in consequence of expansive working, and of the soundness of the method adopted, so far as it goes, for calculating the quantities condensed. The suggestion that the deficiency of water as steam indicated is caused by priming is not sustained by the evidence, for the deficiency is less as the rate of consumption of water per hour is greater; whereas, if it had been caused by priming, the deficiency would have been increased with the rate of consumption.

The results of the performance of another locomotive on the Caledonian Railway, No. 48, of the class of No. 42 just noticed, may be given. This engine was run, March 27th, 1850, with a coke-train up Beattock incline, which is nearly 10 miles long, and of gradients varying from 1 in 80 to 1 in 75. The ascent was performed in twenty-nine and half minutes, and 42.7 cubic feet of water were consumed from the tender. The engine was worked for 3 miles in the first notch, and 6⅞ miles in the third notch, cutting off at 15.7 inches and 11.25 inches, and shutting in for compression at 1.15 inches and 3.01 inches. Adding 1.1 inches of clearance, the volumes of steam admitted per mile were 1809 cubic feet and 1415 cubic feet respectively; and, compressed, 258 feet and 471 feet. The pressures at cutting-off were 50 lbs. and 66 lbs. per square inch; and, for compression, 3 lbs. The quantity of water as steam consumed, as indicated, was 41.09 cubic feet. Then—

	Cubic Feet.
Water consumed from the tanks..................	42.70
Indicated water as steam consumed.............	41.09
Difference, steam condensed...............	1.61, or 3.77 per cent.

The average period of admission was 11½ inches of the stroke of 20 inches, or 57½ per cent. For this admission, the indicated proportion of steam condensed, according to table No. 101, page 474, is 3.15 per cent, whilst the result of the trial shows a loss of 3.77 per cent, showing, though not equality, a near and instructive correspondence.

The foregoing are results of the author's experimental investigations on the working of steam expansively in locomotives, drafted from his paper of 1852, already referred to, as well as from the portion of his work on *Railway Machinery*, published in the course of 1851.[1] These conclusions on the alternate condensation and re-evaporation of steam in cylinders are now universally accepted; but it does not appear that they were fully appreciated for some years after they were published.

Three or four years later, in 1855, M. G. A. Hirn published results of his investigations of the action of the steam-jacket.[2] In this paper he does not make any reference to the action of the material of the cylinder in alternately condensing and re-evaporating steam worked expansively. He simply seeks to establish "that if the steam does not occupy the volume due to its density and pressure, this is due to its being partly condensed into water," and that the steam-jacket, heating the walls of the cylinder, diminishes the quantity of such partial condensation. In his next paper, dated October 29th, 1856, published in 1857,[1] he publishes the results of experimental investigations made by him since the reading of his paper in 1855, in which he, for the first time, apprehends the fact of the alternate condensation and re-evaporation of steam worked expansively in the cylinder, and establishes it in his own manner. He notes that absorption of heat and re-evaporation are continued to the end of the stroke, after the exhaust is opened to the condenser, when expansion and lowering of temperature of the steam in the cylinder are continued. This continuation of the re-evaporative action, which was a very natural deduction, was the only new feature at the time, in M. Hirn's deductions on the alternate condensing and re-evaporative process in the cylinder.

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[1] Whilst the original matter, and the course of investigation originally adopted, have been given, all the calculations and tables, based on the properties of steam, have been made anew and recast, in the light of the more recent experimental investigations of M. Regnault into the constituent heat and density of steam.

[2] "Mémoire sur l'Utilité des Enveloppes à Vapeur," dated April 25th, 1855, published in the *Bulletin de la Société Industrielle de Mulhouse*, 1855; vol. xxvii. page 105.

M. Hirn defined such reactions with a greater degree of precision in 1862.[2] "The steam," he says, having assumed, for example, that it is cut off at one-fifth, "flows from the boiler to the cylinders during one-fifth of the stroke of the piston, from each end of the cylinder. During this part of the stroke the cylinder-cover, the face of the piston, and the corresponding portions of the walls of the cylinder, are necessarily raised to the temperature of the steam, and steam is condensed until this condition is fulfilled. When the steam is cut off and exhaustion begins, the steam falls in temperature, and absorbs heat from the previously heated portion of the cylinder. As the piston advances, the steam parts with heat to the newly-exposed parts of the cylinder, whilst it absorbs heat from the previously heated portions. After passing the middle of the cylinder, the steam meets with the portion towards the other end of the cylinder that had been heated during the previous stroke. When the piston has arrived at the end of the stroke, the steam is exhausted into the condenser, and during this new expansion it absorbs heat from the whole of the surface of the cylinder exposed to it, varying in quantity according to the rapidity of the exhaust, being greater in proportion as the outflow is slower." Here, as in M. Hirn's paper of 1856, there is nothing new, in respect of the subject of the author's investigations, except the statement of the continued absorption of heat and re-evaporation of steam after the exhaust is opened to the condenser.

The existence, then, of alternate condensation and re-evaporation of steam in the cylinder had been experimentally demonstrated by the author many years before M. Hirn had even suspected their existence.

Chief-engineer B. F. Isherwood, of the United States navy, published, in 1863,[3] the results of experiments made by him in 1860, in which he accounts for the inferiority of the quantity of water as steam cut off, measured by the indicator, to the quantity of water delivered to the boiler and passed through the engine. He does so on the principle of the alternate condensation and re-evaporation of steam in the cylinder, the evaporation continuing after the exhaust is opened to the condenser. This explanation is identical with that which had previously been announced, first by the author, and secondly by M. Hirn; and it was published thirteen years after the author had published his own experimental conclusions, establishing the same principles of action, and accounting for the difference of the indicated and measured quantities of water consumed in the same way.[1]

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[1] "Mémoire sur la Théorie de la Surchauffe dans les Machines à Vapeur," in the *Bulletin de la Société Industrielle de Mulhouse*, 1857; vol. xxviii. page 5.

[2] *Exposition analytique et expérimentale de la Théorie mécanique de la Chaleur;* editions of 1862, 1865, and 1876.

[3] *Experimental Researches in Steam Engineering*, vol. i. 1863, page 29.

CHAPTER VI.—BEHAVIOUR OF STEAM IN THE CYLINDER DURING EXHAUST.

According to the received laws of fluid resistance, the exhaust back-pressure should vary as the positive pressure of the steam when cut off or when released, and as the square of the speed of the engine. The following instances, table No. 105, taken from the indicator results of locomotive performance, prove conclusively that the exhaust-pressure measured from the atmospheric line varies as the pressure at the point of cut-off. Each of these couplets of observations was made under the same circumstances.

Table No. 105.—RELATION OF EXHAUST-PRESSURE TO INITIAL PRESSURE IN THE CYLINDER.

Engine and Speed.	Cut-off Pressure.	Exhaust Back-pressure.	Ratio.
C. R., No. 73, 29 miles per hour......	12 lbs. 22 "	5.5 lbs. 9 "	1 to .45 1 to .42
" 32 "	16 " 32 "	6 " 12 "	1 to .38 1 to .38
" 34 "	29 " 36 "	10 " 14 "	1 to .34 1 to .38
" 39 "	23 " 35 "	9 " 14 "	1 to .40 1 to .40
	Exhaust-pressure.		
C. R., No. 33, 36 "	19 lbs.	6.5 "	1 to .34
33½ "	23 "	10 "	1 to .36
" No. 125, 29 "	8 " 28 "	3.5 " 12 "	1 to .44 1 to .44
" " 28 "	10 " 30 "	3.5 " 10 "	1 to .34 1 to .33
E. & G. R., Nile, 41 "	38 " 53 "	5 " 7 "	1 to .13 1 to .13
" 42 "	39 " 52 "	5 " 7 "	1 to .13 1 to .14

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[1] See a paper "On the Behaviour of Steam in the Cylinders of Locomotives during Expansion," by D. K. Clark, in the *Minutes of Proceedings of the Institution of Civil Engineers*, vol. lxxii. p. 275.

With regard to the variation of back exhaust-pressure with speed, an example is reduced from forty indicator diagrams taken from the Nile, E. & G. R., in the course of an express trip. The diagrams have been selected and grouped according to the speeds, being averaged together when the speeds did not differ more than two miles per hour. The cut-off pressures, average back-pressures, and speeds, are given in columns 1, 2, and 3 of the annexed table, No. 106. In the 4th column the actual back-pressures in column 3 are, for the sake of direct comparison, reduced for a common pressure of 40 lbs. at cut-off, in the ratio of the actual cut-off pressures to 40 lbs. The reduced exhaust-pressures are ranged graphically on the diagram, fig. 215, in which the speeds are measured on the base-line AB, from A towards B, and the corresponding back-pressures are plotted on ordinates, and marked by crosses. The curve CA traced through the crosses, represents the back-pressure as it varies with the speed, and it is of a parabolic form, showing that the back exhaust-pressure varies as the square of the speed, other circumstances being the same. The extreme ordinate, BC, measures, by the scale, 9 lbs. of back-pressure, at a speed of 50 miles per hour; at half the speed the measured pressure is only one-fourth, or 2½ lbs. The back-pressures measured from the curve, corresponding to the cut-off pressures, column 2 of the table, are entered in column 5, for comparison with the observed back-pressures, column 4. The agreement is nearly exact.

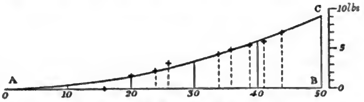
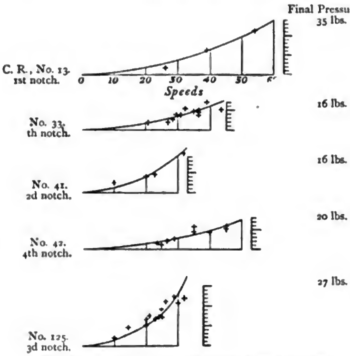


Fig. 215.—Speeds. E. and G. R., Nile:—Curve showing relation of Back Exhaust-pressure to Speed.

Table No. 106.—Nile, E. & G. R.—Back-pressures of Exhaust, in Relation to Speed.

Cylinder 16 inches in diameter, 18 inches stroke; cut-off at 12.3 inches, exhaust at 16.25 inches, wheel, 6 feet.

Speed of Engine. 1	Initial Pressure at Cut-off, per Square Inch. 2	Back Exhaust-pressure. Observed, per Square Inch. 3	Back Exhaust-pressure. Reduced for a common Pressure of 40 lbs. at Cut-off. 4	Back Exhaust-pressure. Measured from the Curve. 5
miles per hour.	lbs.	lbs.	lbs.	lbs.
16	33	0	0	.9
20	25	1	1.6	1.5
24	26	1.5	2.3	2.1
26	38	3	3.2	2.4
34	39	4.1	4.2	4.2
36	35	4.25	4.9	4.7
39	45	6.2	5.5	5.5
41	43	6	5.6	6
44	52	9	7.0	7



Figs. 216.—Relations of Back Exhaust-pressure to Speed.

The evidence of indicator diagrams from the Great Britain and other inside-cylinder locomotives, show the same law of the increase of back exhaust-pressure as the square of the speed.

Outside cylinders, in general, exhibit the operation of the same law of back-pressure. The annexed figs. 216, of plotted observations and parabolic curves, are illustrations in point. No. 13, C. R., must be noted as affording remarkably direct proof of the influence of speed. The three diagrams, figs. 197, 198, and 199, page 462, were taken in succession from No. 13, under the first notch, and in exactly the same circumstances, during an express trip, at speeds respectively of 26, 39, and 54 miles per hour, yielding the results given in the table No. 107:—

Table No. 107.—No. 13, C. R.:—BACK-PRESSURE OF EXHAUST IN RELATION TO SPEED.

Speed, in Miles per Hour.	Indicated Release-pressure, in lbs. per Square Inch.	AVERAGE BACK EXHAUST-PRESSURE, in lbs. per Square Inch.		
		Observed.	Reduced for a Common Release-pressure of 35 lbs. per Square Inch.	Measured from the Curve.
1	2	3	4	5
miles.	lbs.	lbs.	lbs.	lbs.
26	35	1.75	1.75	2.5
39	35	6.00	6.00	5.7
54	32	10.00	10.94	11.0

The nearly equal values of the back exhaust-pressure in the last two columns emphatically affirm the law of the square of the speeds as a measure of back-pressure in outside cylinders steadily at work.

Exceptional Influence of Water in the Cylinder on Back-pressure.

Under special circumstances, the law of the increase of back-pressure with the speed may be modified, as in the cases of C. R., No. 73, No. 124, and No. 127. During three days in April, 1850, on which No. 73 was on trial, the conditions were:—*First day:* lap of valve, 1 inch; lead, 5/16 inch; travel, 4 inches; cut-off, 14 3/16 inches; exhaust, 16 3/4 inches; cylinders, 13 inches in diameter, with an 18-inch stroke; blast-orifice, 3¼ inches in diameter; boiler foul, water muddy. *Second day:* valves altered; lead, $^7/_{16}$ inch; cut-off, $12^{11}/_{16}$ inches; exhaust, 15⅞ inches; blast-orifice, 3¾ inches; boiler foul, as on the previous day. *Third day:* boiler had been run off, and filled with fresh water.

The thick-line diagram, fig. 217, is a copy of one of a number taken from the cylinder during the first day, the beginning of the steam-stroke being towards the right hand. Though liberally aided with lead, the steam did not reach its maximum pressure, 24 lbs., until after one-third of the stroke was described. This result proves a case of extreme priming, and accordingly, when the speed of 41 miles per hour, or 690 feet of piston per minute, was attained, a back-pressure of 11 lbs. was the consequence, or nearly half of the positive pressure. Again, at 15 miles per hour, or 252 feet of piston per minute, there was 13 lbs. of back-pressure, for 30 lbs. positive pressure, as shown by the thin-line diagram. Even after starting, going at 5 miles per hour, the dot-line diagram shows a back-pressure of from 5 lbs. to 10 lbs. Of the whole quantity of water passed through the engine, 24 per cent was proved by measurement to consist of priming. But the back-pressure fell to one-half by substituting fresh water in the boiler for the third day's performance, as shown by the diagram, in dotting, when at the same time the priming was reduced to 11½ per cent of all the water consumed. The table No. 105, page 484, of back-pressures in No. 73, based on the performance on the third day, shows that the back-pressure hardly increased faster than the speed simply.

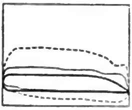


Fig. 217.—C. R., No. 73:—Influence of Water in the Cylinder on Back-pressure.

A special example of the effects of priming in the cylinder of the Great Britain was quoted, page 464, giving proof of the rise of back-pressure—from 15 lbs. to 24 lbs.—in consequence of priming; and it is within the experience of Sir Daniel Gooch that, in similar cases, the exhaust-line has been raised to a level with the steam-line, when the back-pressure amounted to 100 per cent of the positive pressure.

In the diagrams from the Orion, E. & G. R., fig. 203, page 463, it is shown that, by supplying clean water in place of muddy water, the back exhaust-pressure was reduced from 9 lbs. with muddy water to 1½ lbs. per square inch with clean water.

The back exhaust-pressure exhibited by the goods engine, C. R., No. 124, under the first notch, exhausting at 23 inches in a stroke of 24 inches, is shown by the curve, fig. 218, based on the results of thirty-two observations, reduced for 30-lb. release-pressure. The curve is nearly straight, indicating that the back-pressure varied nearly as the speed, and that there was much excess of resistance at the lower speeds from priming or from condensation, or both together. In the course of some experiments with No. 127, an engine of the same class as No. 124, the incidental back-pressure of exhaust was strikingly contrasted for low pressures and for high pressures, and for extreme expansive working. During a preliminary run of 20 miles alone, under the fifth notch, or in mid-gear, cutting off at 4 inches, and exhausting at 14½ inches of the stroke, the indicator diagrams in thick and thin lines, fig. 219, were taken at 17 miles and 29 miles per hour respectively, after 9 miles of the trip had been run, when consequently the engine had got into its regular working condition for that notch. With release-pressures of 24 lbs. and 21 lbs. respectively, the back exhaust-pressures amounted to 14 lbs. and 12 lbs. per square inch. This formidable degree of resistance, in a case where the steam was cut off at one-sixth, is a conspicuous proof of the influence of condensation in the cylinder in aggravating back-pressure:—a condensation which, in the second instance, with a cut-off pressure of 56 lbs., and expansion to nearly three times, amounted to at least 35 per cent of the steam admitted into the cylinder. When the engine was coupled to its train, the back-pressure to a great extent disappeared under longer admissions, and therefore smaller degrees of condensation, as exemplified in the dot-line diagram, fig. 219, taken, at a speed of 23 miles per hour, at the upper end of an incline, after the engine had been well warmed with hard work. Again, when the engine alone with its tender ascended Beattock incline, at speeds of 15 miles and 23 miles per hour, the thick-line and thin-line diagrams, fig. 220, taken at these speeds respectively, show, per 20 lbs. steam cut off, back-pressures of 6 lbs. and 8 lbs. Whereas, in ascending with the train, at 13 miles per hour, in full gear forward, cutting off at 16⅝ inches, with 95-lb. steam-pressure, the back exhaust-pressure, as shown by the diagram in dot-lining, vanished.

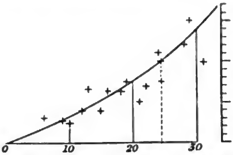


Fig. 218.—C. R., No. 124:—Influence of Water in the Cylinder on Curve of Exhaust-pressure.

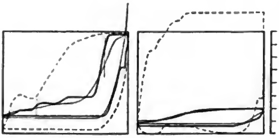


Fig. 219. Fig. 220.
C. R., No. 127:—Influence of Water on Back-pressure.

In inside cylinders insufficiently heated, extra back-pressure, caused by condensation-water, is occasionally apparent immediately after a train is started from a terminus. On the Edinburgh and Glasgow Railway, for instance, in the course of the ascent of Cowlairs incline, on starting from Glasgow, the locomotives, assisted by the stationary engine at the summit, and working under an easy pressure, exhibited a degree of back-pressure which was only abated after a run of several miles had been made with full steam on. With the Brindley, on this railway, of which the cylinders, though inside, were, to be sure, but imperfectly protected, from 7 lbs. to 10 lbs. of back-pressure commonly existed for the first few miles run, and the resistance ultimately fell to a regular pressure of about 2 lbs. per square inch.

Influence of the Period of Release on Back Exhaust-pressure.

The period of release is the interval of space in the cylinder, or the portion of the steam-stroke, during the describing of which by the piston, the port is open to the exhaust.

In all the inside cylinders that were tried, exhaust back-pressure became less as the period of release was increased. This relation is best exemplified in the Great Britain, and may be further elucidated by comparing the relative times of release, as expressed by the angles described by the crank between the points of release and the end of the stroke. Times of release are thus distinguished from periods of release. The comparison is made in the table No. 108, in which the back exhaust-pressures have been reduced for a common release-pressure of 30 lbs. per square inch, and a speed of piston of 600 feet per minute.

Table No. 108.—Great Britain: Back Exhaust-pressure in Relation to Period of Release.

Stroke 24 Inches.

Notch.	Back Exhaust-pressure, at 600 ft. of piston per minute, for 30 lbs. Release-pressure.	Proportional Times of Release, during the steam-stroke, in terms of the corresponding Angles of the Crank.	Periods of Release in percentages of the Stroke.	
	lbs. per square inch.	degrees.	inches.	per cent.
1st notch...........	3.1	33°	3.0	9.4
3d do.	2.3	46	5.0	17.7
5th do.	1.15	59	7.0	27.5

The table indicates that the back exhaust-pressures are generally inversely as the times of exhaust during the steam-stroke.

Still more impressive is the evidence from outside cylinders in favour of earlier release for the shorter admissions. In No. 33, C. R., for the standard speed of 600 feet of piston per minute, whilst the back exhaust-pressure falls one-third as between the 4th and the 5th notches, for the 7th notch the resistance rises. Again, taking together the action in Nos. 124, 125, and 127, C. R., the back-pressure slightly falls, as between the 1st notch and the 3d notch, but it rises towards mid-gear, for which the back resistance is the greatest of all, even with ten times the period of release that belongs to the 1st notch.

Influence of Inside Lead on Back Exhaust-pressure.

In the inside cylinders, well warmed, it does not appear that inside lead had any influence on the back exhaust-pressure. The Hebe, E. & G. R., having less than half the area of inside lead of the Great Britain, exhausted as freely as this engine.

In the outside cylinders, on the contrary, inside lead is very influential in reducing back exhaust-pressure. Take the following comparisons:—

OUTSIDE CYLINDERS, 15 INCHES BY 20 INCHES.

	Inside Lead.	Period of Release.	Back Exhaust-pressure.
C. R., No. 33,	1¼ inches, or $\frac{1}{14.1}$	11.7 per cent.	14.1 lbs.
" No. 51,	1¾ " " $\frac{1}{9.4}$	11 "	7.5 "
" No. 33,	1¼ " " $\frac{1}{14.1}$	14 "	9.9 "
" No. 13,	1 9/16 " " $\frac{1}{11.3}$	13.5 "	4.9 "
" No. 42,	1 9/16 " " $\frac{1}{11.3}$	14 "	5.0 "
" No. 33,	1 5/16 " " $\frac{1}{13.5}$	25 "	10.5 "
" No. 42 (mean of 4th and 5th notches),	1⅝$\frac{1}{32}$ " " $\frac{1}{10.7}$	25 "	5.375 "
C. R., No. 51 (mean of 3d and 4th notches),	1 7/8 " " $\frac{1}{8.7}$	25 "	2.7 "

Here, the better exhausting power of No. 15, having the greatest proportion of inside lead, is conspicuous. No. 33, with the smallest proportion, has obviously the worst exhausting power. In this engine, the valve-gear was originally set with the same lead as No. 42; but, as the gearing had worn, the lead outside and inside was reduced, and the back-pressure was in consequence doubled.

There is, then, a difference between the conditions of inside and outside cylinders, in respect of the necessity for inside lead for ensuring a good exhaust. The following contrasts make this conclusion clear:—

INSIDE AND OUTSIDE CYLINDERS.

	Inside Lead.	Period of Release.	Back Exhaust-pressure.
Hebe, inside cylinder	¾ inch, or $\frac{1}{25}$	15 per cent.	5.8 lbs.
No. 42, outside do.	1 9/16 " " $\frac{1}{11.3}$	14 "	5.0 "
Orion, inside do.	1⅛ " " $\frac{1}{15.7}$	10 "	7.4 "
Nile, do. do.	1 5/32 " " $\frac{1}{15}$	9.7 "	7.5 "
No. 51, outside do.	1¾ " " $\frac{1}{9.4}$	11 "	7.5 "
Gt. Britain, inside do.	1 9/16 " " $\frac{1}{12.5}$	27.5 "	1.15 "
No. 51, outside do.	1 15/16 " " $\frac{1}{8.6}$	28.0 "	2.3 "

In the first contrast, above, the small blast-orifice of the Hebe accounts for the excess of back-pressure, while its lead is not half of that of No. 42. In the second and third contrasts, the inside cylinders exhaust as effectively as the outside cylinders, whilst they have less than two-thirds of the inside lead.

CHAPTER VII.—BEHAVIOUR OF STEAM IN THE CYLINDER DURING COMPRESSION.

The resistance of compression, as represented on the indicator diagram of a non-condensing engine, was found to be measured by the triangular area described by the pencil towards the end of the return-stroke, the base-line of which was to be represented by an extension of the back-exhaust line parallel to the atmospheric line. The compression curve is analogous to the expansion curve, though it is described reversely.

A curve of compression is described equally, whether steam or air be the resisting element. When the steam is shut off from the cylinder, and the piston works in the ordinary atmosphere of the cylinder, the diagram, fig. 221, taken under such circumstances, shows that during the return-stroke, when the line *a b c* is described, the air in the cylinder is exhausted in advance of the piston, moving from *a* until the point *b* is reached, when the valve closes the exhaust. From this point the inclosed air, compressed by the advancing piston, rises in pressure—as indicated by the ascending curve *b c*—till the steam-port is opened for "admission," that is, opened to the valve-chest at the point *c*. Instead of the usual saltation of the pencil, caused by the advent of steam of higher pressure, when steam is there, a descent of the pencil takes place, because the compressed air in the cylinder, now placed in communication with and expanding into the valve-chest and the steam-pipe, falls in pressure in relieving the partial vacuum in these localities—created by the recession of the piston in the cylinder in performing what, if steam were present, would be called the steam-stroke. This partial vacuum is, in fact, indicated for the next "steam-stroke" by the mean dot-line, ranging below the atmospheric line, showing that as the piston advances in returning towards the other end *a* of the cylinder, the pressure continuously falls in consequence of the continuous enlargement of the aggregate capacity of the cylinder, valve-chest, and steam-pipe; until the valve opens the port to the atmosphere through the blast-pipe, near the end of the stroke, when by the influence of external air the pencil rises to the atmospheric line at *a*, the end of the stroke, and equilibrium is restored.

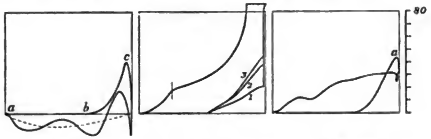


Fig. 221.
C. R., No. 51.—Mid Notch.
37 miles per hour. Steam off.

Fig. 222.
C. R., No. 125.—Mid Notch.

Fig. 223.
E. and G. R., Brindley.
26 miles per hour.

Illustrations of Compression in the Cylinder.

The variation of pressure of steam by compression is, as in the case of expansion, affected by the temperature of the cylinder. As the temperature is then, after a period of exhaustion, at a minimum, partial condensation ensues, particularly in outside cylinders, as the pressure and temperature of steam rise together. Such condensation is shown by some of the low-speed diagrams already figured, where successive condensation-lines of different elevation imply different degrees of condensation. The diagram, fig. 222, from C. R., No. 125, was taken at a low speed, when the compression-lines 1, 2, and 3 successively, were described during three revolutions of the wheel, showing successively higher pressures, and proving less and less condensation as the cylinder became hotter. Diagrams from the Orion show a more prompt ascent of the compression-line at the higher speeds of 30 miles and 40 miles than at 18 miles per hour. Similarly for the Nile and the Brindley, the compression-line has been found to rise much less at the lowest speeds than at the higher speeds. The more rapid rise of compression-pressures at the higher speeds is doubtless in consequence of the hotter condition of the cylinder and the reduction of condensation.

As compression is not directly dependent on the pressure of steam during admission, it occasionally happens, as in fig. 223, that the pressure is raised by compression above the level of the steam-line. The culminating point *a* is reached at the time the port is opened for steam, when the compressed steam falls into equilibrium and the pencil falls to the steam-line.

The area of compression on the indicator diagram is regulated by the total pressure of the exhaust-steam at the closing of the exhaust. It is nearly the same for all speeds, though it increases a little as the speed is increased: the increase being due to higher exhaust back-pressures, which are augmented by the contraction of the outlet as the exhaust-port closes —an operation the reverse of wiredrawing. The increase for higher speeds is also due to the hotter condition of the cylinder. The area of compression depends also on the volume of the total clearance in the cylinder and the steam-port.

CHAPTER VIII.—PRACTICE OF EXPANSIVE WORKING IN SINGLE-CYLINDER STEAM ENGINES.

I. SINGLE-CYLINDER NON-CONDENSING STEAM ENGINES.

Locomotive Engines.

It has been shown that the performance of steam in the cylinders of locomotives was very much affected by the state of protection of the cylinders from external cooling action, and still more by the positive drying or superheating of the steam, whether in the steam-pipes from the boiler passing through the smoke-box, or in the valve-chests and cylinders suspended in the hot atmosphere of the smoke-box, or covered with hot ash and cinders.

Resuming the case of the inside-cylinder locomotives, it was shown that the calculated weight of sensible steam, if taken as saturated, was less at the end of the period of expansion than at the commencement—the point of cut-off. This is an unreal deduction, and, in the light of the ascertained fact of alternate condensation and re-evaporation of steam in the cylinder, it is evident that the steam was, in fact, in a superheated state during admission, and that it behaved as a permanent gas, if not for the whole period of expansion, at least for the greater portion of it. The cylinder having been kept hot,—at least as hot as the steam,—and the steam having been rendered dry and gaseous by the heat of the smoke-box, there was, under the circumstances, no material degree of transference of heat between the steam and the cylinder; and the steam-gas was permitted to expand adiabatically, or in such conditions that heat was neither received by nor given out by the gas, whilst it expanded, the indicator describing an adiabatic expansion-curve.

The work done during expansion is the result of the disappearance of a quantity of heat, which is converted into the work; and, by the abstraction of a given quantity of heat, gaseous steam, as a non-condensing gas, which depends on its specific internal heat only for such a supply of heat, falls more rapidly in pressure and temperature than steam in a state of saturation would do, seeing that this steam has a much greater internal source of heat,—latent heat,—to draw upon than gaseous steam has. In gaseous steam, latent heat is not available for conversion into active work.

To aid in estimating the condition of the steam, suppose that the expanding steam passes during expansion from the gaseous state to a state of saturation, after having consumed its superheat in performing the work of expansion. The weight of steam in the cylinder is, on this assumption, to be measured in terms of the volume and pressure as saturated steam at the end of the expansion. Take the indicated work of the Great Britain, in the first notch. The final weight of steam, as saturated, is (page 471) .5436 lb., occupying the expanded volume 3.357 cubic feet. The total work done by expansion, averaged from the diagrams, is 8659 foot-pounds, which, divided by 772 foot-pounds, the mechanical equivalent of one unit of heat, gives 11.22 units of heat converted into work by expansion. This quantity of heat is, by the assumption, held by the .5436 lb. of steam at the point of cut-off, which, to hold this heat, must be sufficiently elevated above the temperature due to its pressure, as saturated steam. That is to say, a small portion of the heat is held by the steam as saturated, at the initial pressure, in so much as its constituent heat as saturated steam is in excess of that of the expanded steam of lower pressure, as saturated; and the remainder of the heat is held by the initial steam at superheat. The constituent heat of 1 lb. of saturated steam at the final pressure, 68.7 lbs. total, is, by table No. 4, page 29, 1173.4 units; and, at the initial pressure, 94.4 lbs. total, it is 1180.1 units. The difference, 6.7 units for 1 lb., is, for .5436 lb. of steam, ($6.7 \times .5436 =$) 3.64 units, being the fraction of constituent heat set free for expansive working between the higher and the lower pressures. But 11.22 units of heat are converted into work by expansion, and the remainder, (11.22 − 3.64 =) 7.58 units of heat, must be provided as superheat in the steam when cut off. The specific heat of gaseous heat is .475; that is to say, 1 lb. of gaseous steam absorbs .475 unit of heat in being raised 1° F. in temperature, and .5436 lb. absorbs (.475 × .5436 =) .2583 unit of heat for 1° F. rise. Consequently, for the absorption of 7.58 units of heat by .5436 lb. of gaseous steam, it must have acquired (7.58 ÷ .2583 =) 29°.30 F. of superheat or excess temperature above the temperature as saturated steam. The heat converted into work by the expansion of .5436 lb. of steam is, then, as follows:—

Excess of constituent heat as saturated steam, at the initial pressure..	3.64 units.
Superheat at the initial pressure...	7.58 „
	11.22 „

An approximate estimate of the volume of the superheated initial steam may be formed in terms of the absolute temperature of the steam, which, for saturated steam of 94.4 lbs. total pressure per square inch, is (323°.7 (the sensible temperature) + 461° =) 784°.7. The temperature is raised 29°.30 by superheat, to (784.7 + 29.3 =) 814°, and the volume, correspondingly, is expanded in the ratio of 784°.7 to 814°, or of 1 to 1.037. As the volume of 1 lb. of saturated steam of 94.4 lbs. total pressure is 4.58 cubic feet, that of .5436 lb. is (4.58 × .5436 =) 2.491 cubic feet, and the augmented volume by superheat is (2.491 × 1.037 =) 2.584 cubic feet. The actual initial volume of the steam is 2.620 cubic feet, which is greater than the volume now estimated, and the excess volume may be accounted for by the rapidity with which saturated steam commences to expand by superheat: whereas the rate of expansion was taken as uniform in the calculation, seeing that the exact law of the expansion of saturated steam when superheated has not yet been determined.

Following the same line of calculation for the 3d and 5th notches of the Great Britain, the works by expansion are respectively 10,088 foot-lbs. and 11,658 foot-lbs., equivalent to 13.07 and 15.10 units of heat, provided thus:

	3d Notch.		5th Notch.
Excess of constituent heat as saturated steam at the initial pressure........................	4.05 units.	....	3.89 units.
Superheat at the initial pressure.................	9.02 „	....	11.21 „
	13.07 „		15.10 „

The initial weights of steam, taken as identical with the weights of the expanded steam, are .3821 lb. and .2260 lb.; and the volumes of these, as saturated, at the initial total pressures, 87.6 lbs. and 82.4 lbs., are 1.876 cubic feet and 1.177 cubic feet. When expanded by superheat, their initial volumes are estimated to be 1.996 cubic feet and 1.336 cubic feet. But their actual initial volumes are 2.032 cubic feet and 1.295 cubic feet—in the first instance a little more, in the second instance a little less, than the calculated volumes.

The results of these estimates are fairly in accord with the actual circumstances, and they tend to prove a superheated condition of the steam admitted into the cylinder. They are corroborated by independent calculations of the final pressures after the working expansion of the steam, according to the adiabatic ratio of volume and pressure of gaseous steam. The adiabatic expansion of a gas is expressed by the formula $\frac{P}{P'} = \left(\frac{V'}{V}\right)^{\gamma}$, according to which the pressure P in expansion behind a piston varies inversely as the γ power of the volume V, the symbol γ representing the ratio of the specific heat of a gas at constant pressure to the specific heat at constant volume.[1] For gaseous steam, the specific heat at constant pressure is .475, and that at constant volume is .370; and the ratio is as 1.284 to 1, or 1.284. The total pressure of steam expanding on a piston diminishes, therefore, inversely as the 1.284 power of the volume. The pressure falls more rapidly than it does where the steam is maintained at a constant temperature, expanding isothermally, in which case it diminishes simply as the volume inversely. The formula becomes:

Adiabatic Expansion of Gaseous Steam.

$$\frac{P}{P'} = \left(\frac{V}{V'}\right)^{1.284}; \text{ and } P' = P \times \left(\frac{V}{V'}\right)^{1.284} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1).$$

P = the total initial pressure per unit of surface, say, in lbs. per square inch.
P′ = the total final pressure, after expansion, in lbs. per square inch.
V = the initial volume cut off, for expansion, say, in cubic feet.
V′ = the final volume, expanded, in cubic feet.

To calculate the final pressures in the Great Britain, by means of this formula, take the average values for the 1st notch, $V = 2.620$ cubic feet, $V' = 3.357$ cubic feet, $P = 94.4$ lbs. per square inch, and P′ the final pressure.

(1st notch), cut off 67 per cent.....$P' = 94.4 \times \left(\frac{2.620}{3.357}\right)^{1.284} = 94.4 \times .7274 = 68.67$ lbs.

The final pressure, 68.67 lbs. per square inch, here calculated for adiabatic expansion, is identical with the observed total pressure, 68.7 lbs. If the steam had expanded at a constant temperature, the multiplier for the final pressure would have been $\left(\frac{2.620}{3.357} = \right)$ .7805, instead of .7274.

Calculating the final pressures for the 3d and 5th notches, in the same way,

(3d notch), cut off 50 per cent......$P' = 87.6 \times \left(\frac{2.032}{3.062}\right)^{1.284} = 87.6 \times .5906 = 51.74$ lbs.

(5th notch), cut off 29 per cent.....$P' = 82.4 \times \left(\frac{1.295}{2.769}\right)^{1.284} = 82.4 \times .3769 = 31.05$ lbs.

The observed total final pressures were, for the 3d notch, 52.1 lbs. per square inch; and for the fifth notch, 33.2 lbs., with which the calculated adiabatic pressures are nearly identical.

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[1] See *A Manual of Rules, Tables and Data*, by D. K. Clark, page 905.

The pressures, initial and final, of expansion, in the other inside cylinders of the Orion, the Nile, and the Hebe, are similarly in adiabatic relation. These, with those of the Great Britain, calculated adiabatically, are here grouped together for comparison in table No. 109, and the corresponding final pressures calculated isothermally are added in the last column:

Table No. 109.—INSIDE CYLINDERS OF LOCOMOTIVES:—FINAL PRESSURES CALCULATED ADIABATICALLY AND ISOTHERMALLY.

ENGINE. 1	Cut-off at 2	Actual Ratio of Expansion, to point of Exhaust. 3	Total Initial Pressure. 4	Total Final Pressure. Observed. 5	Total Final Pressure. Adiabatically. 6	Total Final Pressure. Isothermally. 7
	per cent.	ratio.	lbs.	lbs.	lbs.	lbs.
Great Britain, 1st notch	67	1.30	94.4	68.7	68.67	72.61
Do. 3d notch	50	1.50	87.6	52.1	51.74	58.40
Do. 5th notch	29	2.14	82.4	33.2	31.05	38.50
Nile........................	67.5	1.30	51.4	37.1	36.88	39.54
Orion	56.7	1.50	61.5	36.5	36.04	41.00
Hebe	44.4	1.64	61	34.0	32.29	37.20

It is observable that, whilst the adiabatic expansive action, indicative of a gaseous or superheated condition of the steam when admitted into the cylinder, is clearly rendered in the foregoing instances, the actual final pressures, for the longer expansions, are sensibly higher than the calculated adiabatic pressures. These slight differences in excess, indicate that the superheat had been exhausted in work, and a state of saturation had been reached, before the expansion was completed. After the state of saturation was reached, the pressure would fall in a less rapid ratio than it would do before, for the reason already explained; and hence the apparent excess. The slight reaction appears in the Great Britain under the 5th notch, cutting off at 29 per cent of the stroke, and in the Hebe, cutting off at 44½ per cent.

The final pressures calculated isothermally, in the inverse ratio of the volumes, in the last column, are several pounds per square inch higher than either the observed or the adiabatic final pressures.

The indicator horse-power of the Great Britain, calculated from the averaged diagrams for each notch, with the quantities of water as steam consumed, are given in the following table, No. 110. For these objects, the calculations have been made on the averages taken out, in table No. 100, page 471. The quantities of water as steam, given in the following table, may be adopted as correctly expressing the consumption of water from the tender, since the Great Britain had been frequently tested for priming, and no priming was detected.

In contrast with this statement, corresponding data for the outside cylinders of Nos. 42 and 48, C. R., are added. For No. 42, the results of the express trip from Glasgow to Carlisle, before analysed (page 481), form the basis of the calculations for the four sections of the trip separately, rendered in table No. 104. For No. 48, the results are based on the observations already made (page 482), with the indicator diagrams.

Table No. 110.—LOCOMOTIVE CYLINDERS:—HORSE-POWER AND WATER AS STEAM CONSUMED. 1850.

Name of Locomotive	Great Britain.			No. 42, C. R.					No. 48, C. R.
	1st notch.	3d notch.	5th notch.	(1)	(2)	(3)	(4)	total	
1. Average speed, miles per hour..........................	34	37	40	29	29	36.9	41.3	35.5	20.3
2. Average speed, feet of piston per minute..........	474	512	563	451	451	573	643	551	316
3. Average period of admission .... per cent	67%	50%	29%	42°/o	52°/o	46°/o	36°/o	42°/o	57.5°/o
4. Average net maximum pressure in cylinders.. lbs.	83	80	81	53	63	62	60	57	62
5. Net pressure at cut-off...lbs.	79.4	72.6	67.4	49.1	61.4	51.0	45.1	49.8	61.0
6. Effective mean pressure, lbs.	68.1	53.5	32.1	31	47.5	38	30	36	51.1
7. Indicator horse-power, I.H.P.	497.8	422.4	278.5	149.7	229.4	233.2	206.6	212.4	172.7
8. Weight of steam at the end of the expansion, per stroke............... lb.	.5436	.3821	.2260	—	—	—	—	—	—
9. Steam detained by compression .... lb.	.0388	.0489	.0624	—	—	—	—	—	—
10. Steam consumed per stroke .... lb.	.5048	.3332	.1636	.1157	.1605	.2184	.1951	.1787	.2384
11. Water as steam consumed per hour...... cu. ft.	231.10	166.00	88.10	71.64	99.36	106.26	84.80	90.35	86.85
12. Water as steam consumed per hour ...... lbs.	14,424	10,360	5,500	4470	6200	6630	5292	5638	5419
13. Water as steam consumed per I.H.P. ... lbs.	28.97	24.52	19.74	29.86	27.02	28.44	25.62	26.54	31.37

In the case of the Great Britain, the consumption of water as steam per horse-power, is reduced almost uniformly as the period of admission is shortened; thus:—

Period of admission .... 67, 50, 29 per cent.
Water consumed per horse-power per hour...... 28.97, 24.52, 19.74 pounds.

showing that the consumption per horse-power would have been still further reduced, for still shorter admissions. In fact, by plotting the above given consumptions as ordinates to a base-line for the stroke, it is seen that, for an admission of 20 per cent, or one-fifth, and an actual ratio of expansion 3.91 reckoned to the end of the stroke, 18½ pounds of steam only would be consumed per horse-power per hour.

The results from Nos. 42 and 48, C. R., show, when plotted, higher rates of consumption of water as steam, averaging about 5½ pounds more per horse-power per hour than those from the Great Britain and the other inside cylinders. Cutting off at 35 per cent, or about one-third, the plotting shows a consumption of 26 pounds of water as steam per horse-power per hour, and, at two-thirds, 34½ pounds; against 20½ pounds and 29 pounds respectively in the Great Britain. The excess of consumption is, no doubt, due partly to the greater proportion of the atmospheric resistance for the lower pressures employed in Nos. 42 and 48, C. R.

Portable Steam Engines.

The trials of portable engines at Cardiff in 1871,[1] afford several examples of steam-jacketed non-condensing engines. A total clearance of 5 per cent has been here assumed for calculation. The engines now selected for comparison are as follows:—

No. 11.	Marshall, Sons, & Co.....	Cylinder	8½	ins. in	diameter,	12 ins.	stroke.
No. 12.	Davey, Paxman, & Co....	,,	8⅝	,,	,,	12 ,,	,,
No. 13.	Brown and May...........	,,	7¼	,,	,,	12 ,,	,,
No. 14.	Tasker......................	,,	9	,,	,,	12 ,,	,,
No. 15.	Reading Engine Works...	,,	8½	,,	,,	14 ,,	,,
No. 16.	Turner.......................	,,	9	,,	,,	12 ,,	,,
No. 17.	Ashley & Co...............	,,	9½	,,	,,	14 ,,	,,

The cylinders of these engines were thoroughly steam-jacketed and lagged. The following leading results, table No. 111, are collected and deduced from the information given in the report. In addition to the average indicator horse-power, as reported, line 9, special horse-power,

Table No. 111.—PORTABLE ENGINES, CARDIFF. HORSE-POWER AND WATER AS STEAM CONSUMED. 1871.

NO. OF ENGINE,..............	11.	12.	13.	14.	15.	16.	17.
1. Average speed, turns per minute,	169	114	139	124	138	179	125
2. Do. feet of piston per minute,	338	228	278	248	322	358	292
3. Average period of admission, %	17%	16%	{10 35} 22.5%	38%	23%	32%	16%
4. Actual ratio of expansion,........	4.8	5.0	{7 2.6} 3.8	2.4	3.8	2.7	5.0
5. Net pressure in boiler, per square inch, ... lbs.	80	80	80	60	80	80	80
6. Net maximum pressure in cylinder, lbs. per square inch,	74–80	75	73	52	72.5	77.2	63
7. Net pressure at cut-off, lbs. per square inch,.....................	63	75	59	48	70	71	57
8. Effective mean pressure,.....lbs.	31.25	33.9	29.2	29.7	37.0	36.24	20.4
9. Reported average indicator horse-power,....................	18.0	13.6	10.1	14.0	20.3	24.8	12.6
10. Actual steam consumed per reported I.H.P.[2],..............	25.90	29.6	31.8	37.9	24.1	27.6	43.2
11. Special indicator horse-power, from sample diagrams, ...	15.0	15.1	10.5	14.1	17.9	25.5	13.3
12. Sensible steam cut-off per special I.H.P.,..................	16.87	14.93	20.52	25.32	18.54	20.08	16.28
13. Sensible steam expanded per special I.H.P., ...	29.82	26.45	28.87	30.27	23.93	22.13	29.98
14. Difference, in parts of expanded steam, per cent	43.4%	43.5%	28.9%	16.3%	22.5%	9.3%	46.0%

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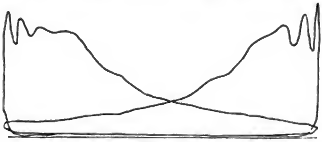
[1] *The Trials of Portable Steam Engines at Cardiff.—Report by the Judges,* 1872.

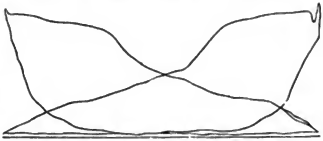
[2] Exclusive of steam supplied to steam-jackets in some cases.

line 11, is calculated from the sample indicator diagrams published in the report. These special horse-powers may not, and do not, exactly tally with the average horse-powers, and the actual quantities of water as steam consumed per indicator horse-power, line 10, are not directly comparable with the sensible quantities of water as steam indicated by the sample diagrams. It is therefore only intended, in the last line, 14, to compare the two steams at the beginning and the end of the expansions in the sample diagrams.

The lowest rate of consumption of water as steam, in line 10, belongs to the Reading engine, No. 15, being 24.1 lbs. per horse-power. The next lowest rate belongs to the Marshall engine, No. 11, or 25.90 lbs. These two engines have the highest speeds of piston, and have high effective mean pressures. They cut off at 23 per cent and 17 per cent respectively; and, looking to the performances of the other engines, it is apparent that increased economy of steam is not effected by cutting off earlier than at one-fourth. The highest consumption of steam, 43.2 lbs. per horse-power per hour, is due to the Ashley engine, No. 17, which cut off at 16 per cent, and made only 20.4 lbs. of effective mean pressure. The comparative economy of cutting off at one-fourth as against one-third is clearly proved by Nos. 15 and 16, the Reading and the Turner engines, in which the effective mean pressures are nearly equal, and the initial pressure is in favour of No. 16; whilst the steams consumed were respectively 24.1 lbs. and 27.6 lbs. per horse-power per hour. The sample indicator diagrams from these two engines are here reproduced for comparison, figs. 224.



Reading Engine, No. 15.



Turner's Engine, No. 16.

Figs. 224.—Portable Engines—Indicator Diagrams.

Separating the results of the diagrams having high maximum pressures nearly alike—between 70 lbs. and 80 lbs. net—from those of low maximum pressures, the respective consumptions per indicator horse-power are as follows. No. 13 is here omitted, as the admissions for the front and the back strokes are excessively unequal.

	Cut-off.	Net Maximum Pressure in Cylinder.	Water per Horse-power.
No. 12	at 16 per cent.	75 lbs.	29.6 pounds.
No. 11	at 17 „	77 lbs.	25.9 „
No. 15	at 23 „	72.5 lbs.	24.1 „
No. 16	at 32 „	77.2 lbs.	27.6 „
No. 17	at 16 „	63 lbs.	43.2 „
No. 14	at 38 „	52 lbs.	37.9 „

Stationary Steam Engines.

The *Porter-Allen Non-condensing Engine* was tested at the Exhibition of the American Institute, New York, in 1870, by Mr. Wellington Lee.[1] The engine is horizontal. The cylinder is not steam-jacketed, except on one side, by the valve-chest. On the exhaust side, a belt is interposed to protect the cylinder externally from the colder exhaust current. The cylinder is covered with boiler-cement cased in wood lagging, except the valve-chest, which is polished. The cylinder-covers are double, and polished. The steam-pipe is coated with boiler-cement, and the steam is supplied from the Allen boiler, in which it is superheated to some extent.

The cylinder is 16 inches in diameter, with a stroke of 30 inches. The total clearance at each end is equal to 1.233 inches in length of the cylinder, making 4.11 per cent of the stroke. The cylinder has four equilibrium slide-valves, two for the admission of steam, and two for the exhaust. The valves are driven from a link worked by one excentric of 2½ inches stroke. The cut-off is variable, but the exhaust-valves are driven from a fixed point at the extremity of the link. The steam-ports are 9 inches by 1½ inches, having about $^{1}/_{15}$th of the area of the piston, and each admission-valve admits steam to the port at four places simultaneously, the movements of the valves being such that they cut off with a rapidly-accelerated motion. The steam-pipe is 6 inches in diameter. Each exhaust-port has a sectional area of 20 square inches. The steam is exhausted at 27½ inches, or 91.66 per cent of the stroke, and compression begins at 2 inches from the end of the return-stroke.

The first trial lasted 11 hours, making 125 turns per minute, under a friction-brake, when the steam was cut off at an average of 11 per cent, or 3.30 inches of the stroke. The total initial and final volumes for expansion, including the clearance, are measured by 4.53 inches and 28.73 inches of the stroke, giving the actual ratio of expansion, 6.34. The average net pressure in the steam-pipe is 73 lbs. per square inch; and in the cylinder the maximum net pressure is 70 lbs. per square inch; and, when cut off, 61 lbs. or 76 lbs. total pressure. The final pressure, at the end of the expansion, is −.455 lb. (below the atmospheric pressure), or 14.345 lbs. absolute pressure. The ratio of the final to the initial pressure is 5.30. A sample indicator diagram is shown in fig. 225. The volume of steam cut off, including clearance, is .5264 cubic foot; the steam weighing, if saturated, .0938 lb. The volume by expansion is 3.342 cubic feet,

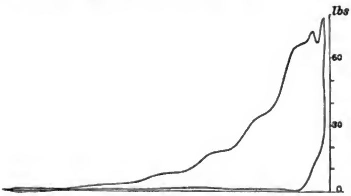


Fig. 225.—Porter-Allen Steam Engine, Non-condensing. First Test Trial: Sample Indicator Diagram. 1870.

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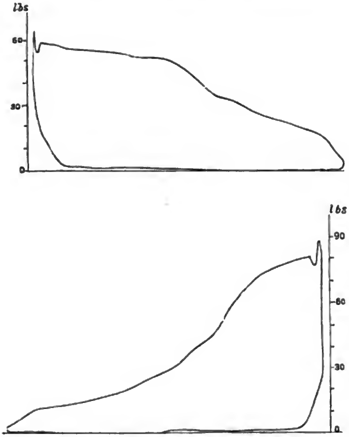
[1] See *Report of Experiments made with the Porter-Allen Engine to determine its Economy and Effective Power*, 2d edition, 1878.

and it weighs .1240 lb.; or .0302 lb. in excess of the initial weight, amounting to 24.3 per cent of the final sensible steam. Here is evidence of alternate condensation and re-evaporation in the cylinder; and that the superheat, which averaged 35°.5 F. in the valve-chest, was much less than what would have been necessary for the prevention of condensation. Mr. Porter was, then, justified in asserting that 50° F. of superheat was found desirable.[1] There is evidence, nevertheless, that the superheat was materially beneficial, in limiting the initial condensation, since the indicated proportion of steam condensed was only 24.3 per cent, against 44 per cent, according to table No. 101, page 474, for nearly the same period of admission, 10 per cent in the table; where no provision was made for superheating the steam, or heating the cylinder.



Figs. 226.—Porter-Allen Engine. Second Test Trial: Sample Indicator Diagrams. 1870.

The indicator diagrams, figs. 226, are samples of diagrams taken from the same engine on another occasion, when it was tested for frictional resistance, at the same test-speed, 125 turns per minute. The steam is cut off at 45 per cent and 23 per cent of the stroke respectively. The excess of the final weight of sensible steam over the initial weight for expansion is 1.8 per cent of the final weight for the first diagram, and 9.7 per cent for the second: showing, in connection with the average result of the cutting off at 11 per cent, percentages of difference and corresponding degrees of alternate condensation and re-evaporation of steam in the cylinder, increasing as the period of admission is shortened. They are here given in comparison with the indicated proportions of steam condensed in unprotected cylinders, using saturated steam, derived from table No. 101:—

	Cut-off.	Condensation.		Cut-off.	Condensation.
	per cent.	per cent.		per cent.	per cent.
Porter-Allen.........	11	24.3	Table No. 101........	10	44
Do.	23	9.7	Do.	25	23.6
Do.	45	1.8	Do.	45	7

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[1] *Proceedings of the Institution of Mechanical Engineers*, 1868, page 56.

The strikingly smaller proportions of condensation in the working of the Porter-Allen engine is a proof of the benefit of superheat in checking condensation during admission, when the results are compared with the proportions arising from unprotected cylinders using saturated steam.

The absolute weight of steam condensed, in relation to the period of admission, may be stated relatively in terms of the periods of admission plus the clearance and the proportional condensation given above, calculated in the same manner as before, for table No. 101, column 7. For the admissions, 11 per cent, 23 per cent, and 45 per cent; or, with clearance added, 15.11 per cent, 27.11 per cent, and 49.11 per cent, the relative absolute weights condensed,—to be directly compared with each other,—are as (15.11 × 24.3%=) 367, (27.11 × 9.7%=) 263, and (49.11 × 1.8%=) 88; or as 1, .72, and .24. These proportions show one-third more steam condensed for 11 per cent than for 23 per cent of admission; and four times as much as for 45 per cent of admission.

The total measured quantity of water as steam consumed was 18,750 lbs., or 300.50 cubic feet: at the rate of 1704.5 lbs., or 27.32 cubic feet per hour. The weight of steam per stroke, as expanded, was stated to be .1240 lb. Deducting the weight of steam shut in and compressed, .0153 lb., the remainder, .1087 lb., is the net sensible quantity consumed per stroke. At the rate of 125 turns per minute, the quantity of sensible steam consumed per hour was (.1087 lb. × 2 × 125 × 60 =) 1630.5 lbs. per hour. This quantity is equal to 95.66 per cent of 1704.5 lbs., the measured quantity of water evaporated, leaving 74 lbs., or 4.34 per cent of water in excess to be accounted for, as follows:[1]—

The temperature of saturated steam of 61 lbs. effective pressure is 308°.4 F., whilst the observed temperature of the initial steam of that pressure was 343°.9 F. in the valve-chest, making 35°.5 of superheat, as before mentioned. Multiplied by .475, the specific heat of gaseous steam, and by .1240 lb., the weight of sensible steam for one stroke, the product, (35.5 × .475 × .1240 =) 2.10 units, is the quantity of superheat. Add the heat freed in expanding from 74 lbs. total pressure to 14.345 lbs. This is calculated by multiplying the difference of the constituent heats of these steams, (1175.4 − 1145.7 =) 29.7 units per pound weight, by .1240 lb., the sensible weight of steam as expanded, making 3.68 units. The sum, (2.10 + 3.68 =) 5.78 units, is the quantity of heat freed for work in expansion. But the total work of expansion is 9411 foot-pounds, equivalent to (9411 ÷ 772 =) 12.19 units of heat. This expenditure is met to the extent of 5.78 units of freed heat, leaving the remainder, 6.41 units, consisting of the heat liberated by the conversion of a part of the expanding steam into water. The work-heat of expansion is, then,—

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[1] It should be stated that the line of analysis here followed, as well as the quantities deduced, are taken up independently of those adopted by the writer of the report, and differ from these in several respects.

By superheat	2.10 units.
By constituent heat set free	3.68 „
By condensation of expanding steam	6.41 „
	12.19 „

These 6.41 units of heat are derived from the latent heat of the steam, of which there are 966 units per pound of the expanded steam of 14.345 lbs. total pressure. To produce 6.41 units, (6.41÷966=) .0066 lb. of steam of this pressure is condensed, and the total steam brought into evidence is, therefore,—

Final expanded steam, sensible	.1240 lb.
Steam condensed, of which the latent heat is converted into work by expansion	.0066 „
Total steam in evidence	.1306 „

Deducting the compressed steam, .0153 lb., the remainder, (.1306 −.0153=) .1153 lb., is the weight of steam consumed per stroke, including the steam condensed for work by expansion: equivalent to (.1153 lb. × 2 × 125 × 60=) 1729 lbs. per hour. This quantity is 24.5 lbs., or 1.44 per cent in excess of the measured quantity of water evaporated, namely, 1704.5 lbs.

The slight difference in excess may have arisen from errors of observation, and from want of allowance for the volume of the piston-rod. The evidence derivable from the indicator diagrams proves that all the water passed from the boiler into the engine can be fully accounted for as steam.

For comparison, the terminal pressures, at the end of the expansion, as calculated under other assumed conditions, are subjoined:—

			per square inch.
Total terminal pressure, observed			14.34 lbs.
Do.	do.	for equal weights of steam	11.00 „
Do.	do.	isothermally (in hyperbolic ratio)	8.57 „
Do.	do.	adiabatically	7.08 „

The effective mean pressure in the cylinder is 17.34 lbs. per square inch, on an area of 201.06 square inches; and at a speed of (2.5 × 2 × 125=) 625 feet of piston per minute, it makes 66.05 indicator horse-power. The steam consumed according to the evidence of the indicator diagram is at the rate of (1729 lbs.÷6605=) 26.18 lbs. per horse-power per hour; whilst the measured quantity of water consumed was at the rate of (1704.5÷66.05=) 25.81 lbs. per horse-power per hour.

The wiredrawing exhibited by the indicator diagrams, figs. 225 and 226, shows the following reductions from the maximum pressure of admission to the pressure at cut-off:—

Admission.	Maximum pressure.	Cut-off pressure.	Difference.	
11 per cent	70 lbs.	61 lbs.	9 lbs., or 13	per cent.
23 „	58.5 „	49 „	9.5 „ or 16	„
45 „	81 „	71 „	10 „ or 12.3	„

These reductions of pressure by wiredrawing are considerably less in proportion than those of the Great Britain and other engines, given in table No. 98, page 459. The less degree of wiredrawing is due to the short steam passages and the multiplied entrances for steam to the cylinder provided by the slide-valves. The relatively rapid movement for cutting off, also, may have contributed to reduce the degree of wiredrawing.

Other instances of non-condensing performance are given under the next heading.

II. SINGLE-CYLINDER CONDENSING STEAM ENGINES.

An *Allen Engine*, at Pooley's Mill, Manchester, of construction similar to that of the Porter-Allen engine, above described, with the addition of a steam-jacket and a condenser, may be noticed in contrast with the non-condensing engine.[1] The engine is horizontal, condensing, and the cylinder is thoroughly steam-jacketed. The cylinder is 18 inches in diameter, having a stroke of 30 inches. It makes 120 turns per minute, with a speed of piston of 600 feet per minute. In the indicator diagram, fig. 227, the steam is cut off at 10 per cent, or 3 inches of the stroke, making, with a clearance of 6 per cent, or 1.8 inches, a total admission of 16 per cent, or 4.8 inches. In expanding to a point at 90 per cent, or 27 in. of the stroke, where the exhaust opens, making a total expanded volume, 96 per cent, or 28.8 inches of the stroke, the actual ratio of expansion is (96÷16=) 6. The total initial and final pressures of expansion are 55 lbs. and 10 lbs. per square inch, having the ratio 5.5. The weight of steam cut off is .0946 lb., and expanded .1142 lb., making a difference of .0196 lb., or 17 per cent of the final sensible steam.

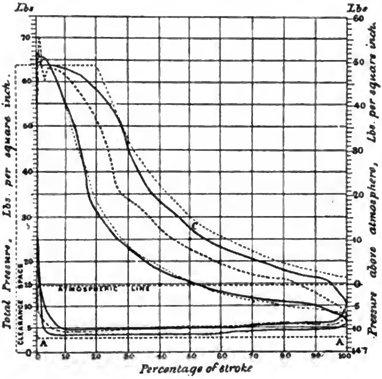

Fig. 227.—Allen Steam Engine: Indicator Diagrams. AA, vacuum line.

The total work of expansion between 10 per cent and 90 per cent of the stroke, is 10,109 foot-pounds, equivalent to (10109÷772=) 13.10 units of heat. This quantity is provided for in part by the heat freed by the fall of pressure from 55 lbs. total to 10 lbs. total per square inch, or the difference of the constituent heats per pound of weight, (1169.0−1140.3=) 28.7 units, multiplied by .1142 lb., the sensible expanded weight, making 3.28 units. The remainder, (13.10−3.28=) 9.82 units of work-heat, is supplied by the latent heat of steam condensed during expansion. The latent heat of steam of 10 lbs., the terminal expansion-pressure, is 978.4 units per pound, and the weight of steam so condensed, for expansive work, is (13.10÷978.4=) .0134 lb. This is to be added to the final sensible steam, .1142 lb., making up .1276 lb. of steam in evidence at the end of the expansion. The compressed steam, .0044 lb., calculated for a total exhaust back-pressure of 3½ lbs., is to be deducted, and the remainder, (.1276−.0044=) .1232 lb., is the weight of steam consumed per stroke, equivalent to (.1232 lb. × 2 × 120 × 60=) 1774 lbs. of steam per hour.

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[1] For a detailed description and illustration of the engine, see a paper by Mr. Charles T. Porter, "On the Allen Engine and Governor," in the *Proceedings of the Institution of Mechanical Engineers*, 1868; page 50.

The total back-pressure on the piston is 4 lbs. per square inch, and the effective mean pressure is 18.2 lbs., making 84 indicator horse-power. The weight of steam proved by the indicator is (1774÷84=) 21.12 lbs. per horse-power per hour, which does not include the weight of steam condensed in the jacket.

Table No. 112.—Results of Performance of Porter-Allen and Allen Steam Engines.

	Porter-Allen, Non-condensing.	Allen, Condensing.
Working condition	superheated steam.	jacketed cylinders.
Diameter of cylinder	16 inches.	18 inches.
Length of stroke	30 inches.	30 inches.
Clearance	1.233 ins., 4.11 per cent.	1.8 ins., 6 per cent.
Turns per minute	125 turns.	120 turns.
Speed of piston, in feet per minute	625 feet.	600 feet.
Cut off at	11 per cent.	10 per cent.
Exhaust at	91.66 per cent.	90 per cent.
Actual ratio of expansion	6.34	6
Total initial pressure, at cut-off	76 lbs.	55 lbs.
Total final pressure, expanded	14.34 lbs.	10 lbs.
Ratio of pressures	5.30	5.50
Weight of initial sensible steam	.0938 lb.	.0946 lb.
Weight of final sensible steam	.1240 lb.	.1142 lb.
Difference	.0302 lb.	.0196 lb.
Per cent of final weight	24.3 per cent.	17 per cent.
Weight of net steam consumed per stroke	.1153 lb.	.1232 lb.
Weight of final sensible steam consumed per horse-power per hour, measured direct from the indicator diagram	24.69 lbs.	21.12 lbs.
Weight of water as steam actually consumed per horse-power per hour	25.81 lbs.	—
Indicator horse-power	66.05 H.P.	84 H. P.

This calculation of the weight of steam consumed brought into evidence by the indicator, has been made without any allowance for heat communicated by the jacket during expansion. It has been assumed that the heat absorbed by the material of the cylinder during the first part of the expansion is counterbalanced by the heat given out by it to the steam during the last part of the expansion.

The foregoing results, deduced from indicator diagrams, of the Allen condensing engine are compared with those from the Porter-Allen engine, page 505, in the table No. 112.

It is apparent that the characters of the expansion-lines are very different:—proceeding, in the one case, from superheated steam; in the other, from saturated steam, aided by a steam-jacket. Cutting off at 11 per cent and 10 per cent of the stroke respectively, the expansion-lines show alternate condensations and re-evaporations of 24.3 per cent and 17 per cent of the steams. Again, the steam-jacket, in the second case, deals, in presence of the condenser, with a greater range of temperature than the superheat of the non-condensed steam, in the first case; and it is to be inferred that the steam-jacketed cylinder displayed a greater degree of efficiency than the cylinder with superheated steam, whilst both of them are clearly superior to an unprotected cylinder with saturated steam.

The total final pressure of expansion, in the Allen engine, calculated isothermally, in the hyperbolic ratio, would be (55 lbs. ÷ the actual ratio 6 =) 9.17 lbs. per square inch, or .83 lb. less than the observed final pressure, 10 lbs. The light hyperbolic line on the diagram, fig. 227, shows a satisfactory agreement with the actual curve of expansion:—although a little fuller at the upper part, and a little more hollow at the lower part.

The steams brought into evidence by two other indicator diagrams from the same cylinder, shown in outline, in fig. 227, are here stated, with the results already ascertained, thus,—

Period of admission,	10	per cent.	Steam per horse-power per hour,		21.12	pounds.
Do.	20	,,	Do.	do.	20.33	,,
Do.	26	,,	Do.	do.	19.92	,,

It appears that the rates of consumption vary inversely with the periods of admission; suggesting that there was less economy in cutting off earlier than at one-fourth of the stroke.

The *Crossness Rotative Pumping Engines*,[1] four in number, have each a 48-inch cylinder, of 9 feet stroke, thoroughly steam-jacketed. They make 11 turns per minute, with an average speed of piston of 198 feet per minute. The total clearance at each end of the cylinder is 2.6 per cent of the stroke. The mean periods of admission at the top and bottom of the cylinders of the respective engines, according to sample indicator diagrams, of which one set is represented in fig. 228, are 13 per cent, 13¾ per cent,

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[1] See Mr. Gilbert Hamilton's paper, "Description of the Crossness Pumping-engines for the Metropolitan Main Drainage Works," in the *Proceedings of the Institution of Mechanical Engineers*, 1867; page 236.

13⅛ per cent, and 18½ per cent, for which the actual ratios of expansion, reckoned to the end of the stroke, are 6.60, 6.30, 6.50 and 4.90; reckoned to 90 per cent of the stroke, they are 6, 5.66, 5.90, and 4.39. The mean total initial pressures for expansion are 46.5 lbs., 42.75 lbs., 50.5 lbs., 48.5 lbs., and the total final pressures at 90 per cent are 9.12 lbs., 7.75 lbs., 10 lbs., 11 lbs. The ratios of the initial to the final pressures are 5.10, 5.51, 5.01, and 4.41. The mean weights of sensible steam, cut off for expansion, are 1.89 lbs., 1.92 lbs., 2.16 lbs., 2.79 lbs.; and at 90 per cent of the stroke, 2.47 lbs., 2.18 lbs., 2.81 lbs., and 3.04 lbs. The differences, or excess weights of final steam, are 23.5 per cent, 12 per cent, 23.1 per cent, and 8.2 per cent of the final weights. These excess weights, indicating alternate condensation and re-evaporation in the cylinder, are much less than what take place in unprotected cylinders, according to table No. 101, page 474; and their smaller proportions are evidence of the benefit derived from the steam-jackets.

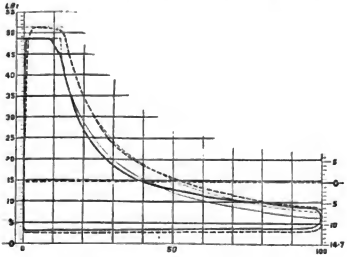

Fig. 228.—Crossness Pumping Steam Engines: Indicator Diagrams, from top and bottom of cylinder.

The net indicated weights of steam consumed per stroke, after deducting steam compressed, are 2.44 lbs., 2.14 lbs., 2.78 lbs., and 3 lbs. To these should be added the weights of steam condensed for expansion-work. Dividing the total work, in foot-pounds, done by expansion to the end of the stroke by 772, the quotients are 291.1, 253.1, 314.3, and 341.7 units of heat converted into work by expansion, produced in part by the free heat disengaged by the expanding steam due to the difference of the constituent heats of the initial and final steams; and, in part, by the heat disengaged by condensation of the expanding steam. The differences of the constituent heats, multiplied by the respective weights of final sensible steam, give the free heats disengaged, 65.95, 61.04, 75.87, and 75.40 units; which, being deducted from the respective quantities of heat converted into work by expansion, give the remainders, 225.15, 192.06, 238.43, and 266.3 units of heat, drawn from steam condensed during expansion, and converted into work. These, being divided by the respective latent heats of the final sensible steams, give the weights of steam so condensed, namely, .229 lb., .195 lb., .244 lb., and .273 lb. Adding these weights to the weights of sensible final steam respectively, and deducting the weights of steam reserved by compression, namely, .029 lb., .030 lb., .029 lb., and .036 lb., the ultimate remainders, 2.670 lbs., 2.345 lbs., 3.025 lbs., and 3.277 lbs., are the weights of steam consumed for one stroke, brought into evidence by the indicator. Multiplied by (11 strokes × 2 × 60 minutes =) 1320, the products, 3524 lbs., 3095 lbs., 3993 lbs., and 4260 lbs., are the weights of steam per hour, proved by the indicator; and these, divided by the respective indicator horse-powers, 187.3 H.P., 168.8 H.P., 200.8 H.P., and 254 H.P., give the weights of steam 18.82 lbs., 18.34 lbs., 19.89 lbs., and 16.77 lbs., per indicator horse-power. Arrange these weights consumed in the order of the corresponding periods of admission, thus:—

Cut off at	13,	13⅛	13¾	18½ per cent
Calculated indicated weight of steam consumed per horse-power..........	18.82	19.89	18.34	16.77 lbs.

The first and second periods of admission are so nearly identical that the mean of the steams may be taken for comparison. It is clear that the consumption of steam diminishes as the period of admission is lengthened from 13 per cent to 18½ per cent; and it may be presumed that the consumption would be still further reduced for longer periods of admission, up to probably 25 per cent of the stroke, for which, by plotting the given results, the consumption of steam is found to be 15½ pounds per indicator horse-power per hour.

That these quantities of steam, deduced from the evidence of the indicator diagrams, represent at least approximately the actual consumptions, is corroborated by the fact that the quantity of coal consumed per indicator horse-power was 2.2 lbs.; seeing that the average consumption of steam as above calculated is 18.45 lbs. per indicator horse-power, which, divided by 2.2, gives 8.39 lbs. of water evaporated per pound of coal,—a reasonably good proportion.

The four *early pumping engines of the East London Water-works* were tested by Mr. C. Greaves[1] for long periods. The engines are "Cornish:" that is to say, single-acting high-pressure expansive condensing engines, working single-acting pumps through the medium of a beam. The steam is admitted and cut off by double-beat valves, and expanded on the upper side of the piston for the "indoor-stroke" or down-stroke, whilst the lower face of the piston is open to the condenser. When the indoor-stroke is completed the expanded steam above the piston is freely admitted and exhausted to the lower side of the piston, which is closed to the condenser, and the piston, the two faces of which are in equilibrium, in the spent steam, quietly ascends to the top of the cylinder. When it arrives near the top the exhaust from the upper side is closed, and the remaining steam is compressed, whilst the exhausted steam below the piston is discharged into the condenser.

The walls of the cylinders are cased in steam-jackets inclosed in an outer casing of fine ashes, 12 inches thick. The covers are not jacketed, but they and the pipes are well encased. The jackets are filled with steam of 35 lbs. effective pressure per square inch—the pressure in the boilers. The steam-valves are double-beat valves.

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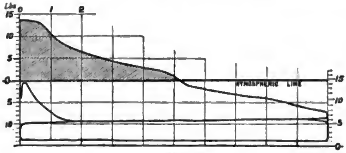
[1] See Mr. Greaves' paper on Cornish Pumping Engines in the *Proceedings of the Institution of Mechanical Engineers*, 1862; page 147.

Name of Engine..........	AJAX.	CORNISH.	WICKSTEED.	VICTORIA.
Diameter of cylinder..............	72 in.	80 in.	90 in.	100 in.
Average length of stroke.........	9 ft. 7½ in.	9 ft. 9 in.	10 ft. 7 in.	11 feet.
Total clearance at the top, in parts of the stroke........	3.67%	3.5%	3.25%	3.33%

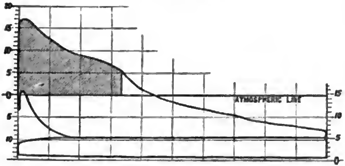
Five pairs of indicator diagrams, figs. 229 and 230, were taken from the 72-inch cylinder for various cut-offs of from ½ stroke to ⅙, doing the same duty at the same speed, 7 strokes per minute. Each pair of diagrams was taken, one from the upper side, and one from the lower or exhaust side. The indoor-strokes were made at speeds of from 500 to 600 feet per minute. For instance, in cutting off at one-fourth, a 10-feet stroke is frequently made in the course of one second, at a mean speed of 600 feet per minute; and an 11-feet stroke in 1¼ seconds, at a mean speed of 528 feet per second.

Select for analysis the work of the 72-inch cylinder with a stroke of 9.625 feet, cutting off at one-fourth—25 per cent. The clearance is 3.67 per cent of the stroke, and the actual ratio of expansion, reckoned to the end of the stroke, is (103.67 ÷ 28.67 =) 3.62. The total maximum pressure during admission is 34¼ lbs. per square inch; the total pressure at the cut-off is 23¼ lbs., and at the end of the stroke 6¾ lbs., the ratio of the pressures being 3.44. The weight of sensible steam cut off is 4.564 lbs., and expanded, 5.163 lbs., the difference, .599 lb., being 11.6 per cent of the final weight of sensible steam.

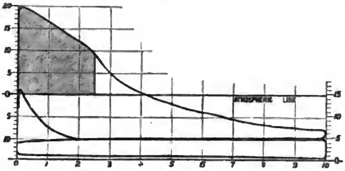
The total work by expansion for one stroke is 344,850 foot-pounds, equivalent to 446.7 units of heat. Of this quantity of heat 95.53 units are supplied as heat set free as the difference of the constituent heats of steams of the initial and final pressures in expansion, and the remainder, 351.17 units, is disengaged from .3543 lb. of steam of the final pressure, condensed during expansion. The total steam in evidence is the sum of this weight condensed, and 5.163 lbs. that of the final sensible steam; or 5.517 lbs. Deducting 59 cubic feet of steam detained, at a total pressure of 4½ lbs., and compressed, .737 lb. in weight, the remainder, 4.780 lbs., is the weight of steam consumed within the cylinder per stroke, as evidenced by the indicator. The steam condensed in the jacket was returned direct to the boiler, and is estimated by Mr. Greaves, from the results of observations on the 90-inch cylinder engine, as 4 per cent of the steam consumed within the cylinder. Making an addition for such steam, the weight consumed per stroke amounts to 4.971 lbs. per stroke; equivalent to (4.971 lbs. × 7 strokes × 60 minutes =) 2088 lbs. per hour. The horse-power, calculated for an effective mean pressure of 15 lbs. per square inch, is 124.6 horse-power; and the quantity of steam, by the indicator, consumed per horse-power per hour is (2088 ÷ 124.6 =) 16.76 lbs. The weight of water as steam recorded as actually consumed over a long period per indicator horse-power is 20.72 lbs. The calculated weight, 16.76 lbs., is 23 per cent less than the observed actual consumption of water for a long period. Such direct comparison is of doubtful use, for there were unmeasured contingencies in the case. Of course, the condensing action of the unjacketed cover of the cylinder made its impression on the performance. Nevertheless, the proportion of the weight of expanded steam condensed was but 11½ per cent, whilst, by table No. 101, page 474, 23½ per cent, or twice as much, was condensed in unprotected cylinders. Besides the likely causes of accidental increase of consumption during a long period of continuous work—as leakage from safety-valves and gauge-cocks, and extra steam used in starting every day—the average movement of the piston for seven double-strokes per minute was only (9.625 feet × 2 × 7 =) 135 feet per minute. The down-strokes, at the rate of, say, 600 feet of piston per minute, were made in (9.625 × 7 ÷ 600 =) .112 minute; and the seven up-strokes, including the stationary intervals, were made in (1 − .112 =) .888 minute, at an average speed of (9.625 × 7 × 1 ÷ .888 =) 76 feet per minute. The extreme slowness of the upward movements, with the intervals of repose at the end of each up-stroke, were the occasion of much condensation, which could not all be brought into evidence by the indicator.



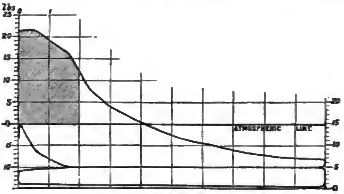
Cut-off at ½.



Cut-off at 1/3.



Cut-off at ¼.

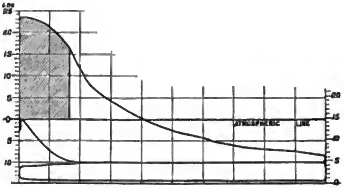


Cut-off at 1/5.

Figs. 229.—East London Water-works: Indicator Diagrams from 72-inch Cylinder.

The results of the foregoing and similar analyses of the performances of the 72-inch cylinder engine Ajax, cutting off at various parts of the stroke, and of the three other engines already named, are given in table No. 113, pages 512, 513.

An instructive contrast is afforded by the five pairs of diagrams, taken from the 72-inch cylinder, combined in fig. 231, cutting off at from one-half to one-sixth of the stroke, showing the same effective mean pressure, with the same speed, and doing the same duty. The calculated quantities of water as steam consumed per indicator horse-power, line 31, show that a period of admission of one-fifth is the most economical; that there is little difference between one-fifth and one-sixth for economy; and that the consumption increases with the period of admission beyond one-fifth. The very small difference between one-fifth and one-sixth is sufficiently explained by the diagrams, which are nearly identical in form, and in which it is seen that by the wiredrawing of the steam in cutting off at one-fifth, the admission of steam virtually ceases at one-sixth of the stroke. The greater efficiency of one-fifth and one-sixth compared with the larger fractions of admission, is due mostly to the higher initial pressures and the greater quantities of work done by expansion, shown in line 15; and in part also to the freer admission of steam, for which the admission-line is full and round, in the shorter cut-offs, and is, on the contrary, hollow in the longer cut-offs. The wiredrawing action is, it is seen in line 10, very considerable, notwithstanding the prompt action of the double-beat valves, which are 16 inches in diameter. The diagrams, fig. 232, of the lift of the steam-valve of the 72-inch cylinder were traced by the valve itself, showing the actual points of cut-off on the base-line of the figures, the length of the base-line representing the stroke. The vertical scale of the diagrams



Cut-off at 1/6.

Fig. 230.—East London Water-works: Indicator Diagrams from 72-inch Cylinder.

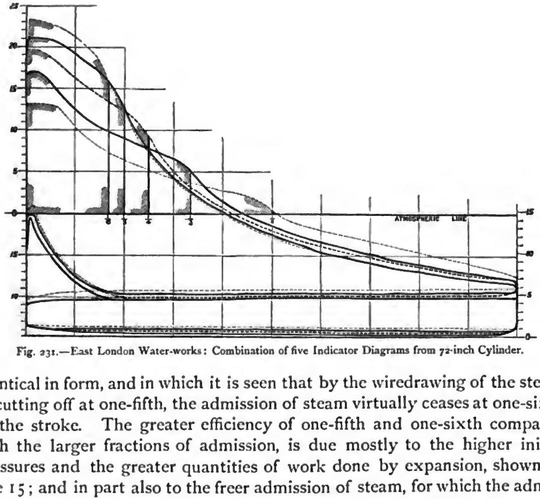


Fig. 231.—East London Water-works: Combination of five Indicator Diagrams from 72-inch Cylinder.

is full size, showing the actual lifts, which were—for an admission of half-stroke, 1.88 inches; of one-third stroke, 1.80 inches; of one-fourth stroke, 1.74 inches; of one-fifth stroke, 1.69 inches; and of one-sixth stroke, 1.53 inches. These diagrams prove that the steam was admitted freely and cut off promptly; and yet the proportions of wiredrawing, line 10, are very considerable, in comparison with those of crank-engines. The initial velocity of the piston, uncontrolled by a crank, was of course much greater than that for a crank-engine; and as the average velocity of the piston in making the indoor-stroke is, as before noticed, from 500 feet to 600 feet per minute, the velocity may have been twice as much in the early part of the stroke. Hence, probably, the excessive wiredrawing, the proportion of which increases with the period of admission.

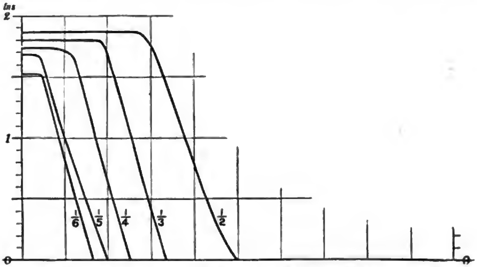


Fig. 232.—East London Water-works:—Diagrams of the Lift of Steam Valve of 72-inch Cylinder.

For practical reasons, says Mr. Greaves, a period of admission of one-fourth is preferred for regular work. For shorter periods, the stress on the machinery becomes excessive. Though an admission of one-fourth does not give the lowest calculated consumption of steam in the 72-inch cylinder, it gives, in the 90-inch cylinder, the lowest measured consumption of all. It may be taken that, in the engines under notice, an admission of about

Table No. 113.—East London Water-works: Pumping Engines—Performance of the Engines.

1. Name of Engine	Ajax.					Cornish.	Wick-steed.	Vic-toria.
2. Diameter of cylinder	72 inches					80 in.	90 in.	100 in.
3. Stroke of cylinder	9.625 feet					9.75 ft.	10.58 ft.	11 ft.
4. Period of admission	½	⅓	¼	⅕	⅙	⅓	¼	¼
5. Actual ratio of expansion	1.93	2.80	3.62	4.38	5.10	2.81	3.66	3.65
6. Total maximum pressure in cylinder lbs.	27.75	31.5	34.25	34.25	37	27	36	40
7. Total initial pressure at cut-off lbs.	15	19.75	23.25	27.25	30.5	17.5	23	29
8. Total final pressure lbs.	7.25	7	6.75	6.25	6.33	7	6.25	7.5
9. Ratio of pressures, Nos. 7 and 8	2.07	2.82	3.44	4.36	4.82	2.50	3.66	3.87

Table No. 113 (*continued*).

NAME OF ENGINE…….		AJAX.					CORNISH.	WICKSTEED.	VICTORIA.
10. Wiredrawing, difference of Nos. 6 and 7…………	p.cent.	46%	37%	32%	20%	18%	35%	36%	27%
11. Weight of sensible steam cut off……….	lbs.	5.652	5.045	4.564	4.375	4.179	5.602	7.457	12.220
12. Weight of sensible steam expanded…….	lbs.	5.501	5.332	5.163	4.768	4.824	6.660	8.101	12.246
13. Difference of Nos. 11 and 12…………..	lbs.	−.151	.287	.599	.393	.645	1.058	.644	.026
14. Difference per cent of No. 12………………….		−2.7%	5.4%	11.6%	8.2%	13.4%	15.9%	8%	.21%
15. Total work by expansion, in heat-units………		273.10	381.00	446.70	497.50	539.10	423.40	751.70	1130.00
16. Supplied partly as constituent heat of steam freed by expansion…………..	units.	58.31	82,11	95.53	100.60	116.30	89.91	157.20	254.60
17. and partly as heat of steam condensed for work by expansion………………	units.	214.79	298.89	351.17	396.90	422.80	333.50	594.50	875.4
18. Weight of steam so condensed…………	lbs.	.217	.302	.354	.399	.426	.337	.598	.886
19. Total weight of steam in evidence by indicator, per stroke, sum of Nos. 12 and 18….	lbs.	5.718	5.634	5.517	5.167	5.250	6.997	8.699	13.132
20. Steam retained by compression……….	lbs.	.830	.814	.737	.737	.737	.635	1.451	1.512
21. Net steam consumed, by the evidence of the indicator……….	lbs.	4.888	4.820	4.780	4.430	4.513	6.362	7.248	11.620
22. Estimated steam condensed in jacket, per stroke………………	lbs.	.195	.177	.191	.177	.180	.255	.290	.465
23. Total calculated net weight of steam consumed per stroke, sum of Nos. 21 and 22	lbs.	5.083	5.008	4.971	4.607	4.693	6.617	7.538	12.085
24. Water as steam consumed per stroke from the boilers for long periods………………	lbs.	—	—	6.15	—	—	7.61	9.97	14.53
25. Difference of Nos. 23 and 24…………	lbs.	—	—	1.18	—	—	.99	2.43	2.45
26. Difference per cent of No. 24….	p. cent.	—	—	19.2%	—	—	13%	24.4%	16.9%
27. Effective mean pressure…………………	lbs.	15	15	15	15	15	14.38	15.58	16.58
28. Strokes per minute………		7	7	7	7	7	7	6	5
29. Average speed of piston per minute…	feet.	135	135	135	135	135	136.5	127	110
30. Indicator horse-power, H.P.		124.6	124.6	124.6	124.6	124.6	149.5	190.7	217.0
31. Calculated water as steam consumed per H.P. per hour……..	lbs.	17.14	16.88	16.76	15.52	15.82	18.59	14.23	16.70
32. Water as steam actually consumed per H.P. per hour for long periods………………	lbs.	—	—	20.72	—	—	21.38	18.82	20.08
33. Difference of Nos. 31 and 32..	p. cent.	—	—	19.2%	—	—	13%	24.4%	16.9%
34. Coal consumed per H.P.……………….	lbs.	—	—	2.07	—	—	2.14	1.88	2.01

one-fourth, gives probably the highest degree of efficiency of steam-power, although, according to the evidence of the calculated steams per horse-power, line 31, a latitude of between one-fifth and one-fourth of the stroke for admission may be allowed in pursuance of the most economical result. There is a considerable difference, line 25, between the quantities of water as steam brought into evidence by the indicator, line 23, and the water actually consumed, line 24. The difference may be due partly to the diverse origin of the two quantities, of which the second, in line 24, was measured for long periods, and partly to the incomplete jacketing in steam,—the walls only of the cylinder having been steam-jacketed.

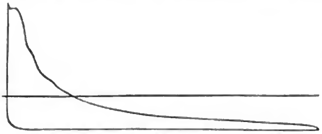
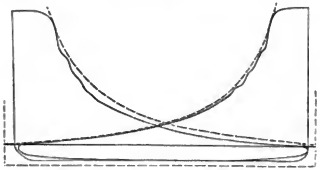
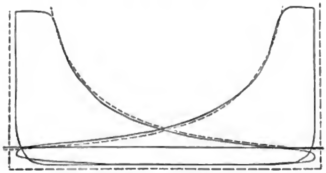


Fig. 233.—Sulzer Steam Engine: indicator diagram.

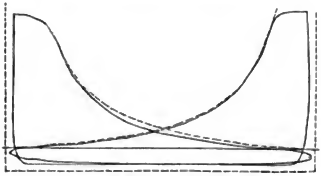


Reynolds-Corliss Engine: indicator diagrams.

A *steam engine, of a modified Corliss type*, by Messrs. Sulzer Brothers, Winterthur, was exhibited at Vienna in 1873. The cylinder was 17¾ inches in diameter, with a stroke of 3 feet 5⅜ inches; steam-jacketed, and thoroughly lagged with cement, felt, and wood. A similar engine, of which a sample indicator diagram is shown, fig. 233, cutting off at 4 per cent, was said to have consumed 18¾ lbs. of water as steam per indicator horse-power per hour. On a pair of 200 horse-power engines of this class, a consumption of less than 20 lbs. of water per horse-power per hour was guaranteed. Professor Lindé tested one such engine, with the following average results of a four-days trial.[1] Making 40 turns per minute, the actual ratio of expansion was 10, with 70-lb. steam, and a pressure in the condenser of .27 lb. Indicating 395 horse-



Harris-Corliss Engine: indicator diagrams.



Wheelock Engine: indicator diagrams.

Figs. 234.—Test-trials, by J. W. Hill.

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[1] See *Report of the American Commission on the Vienna Exhibition*, edited by Professor Robert H. Thurston; vol. iii. page 23.

power, the consumption of water as steam was 19.6 lbs. per horse-power per hour. Evidence is wanting to show whether such a large measure of expansion as 10 was necessary for securing the most economical result.

Three *automatic cut-off engines* were subjected to trials, conducted by Mr. J. W. Hill, at Cincinnati, in 1880.[1] These engines were the Harris-Corliss, the Reynolds-Corliss, and the Wheelock engines, all fitted with liberating valve-gear: the first and the second having the original Corliss valves and gear, with vacuum dashpots, and with special improvements, and the Wheelock engine having a system of taper-plug valves, placed in one plane below the base of the cylinder, with weight dashpots. The comparative tardiness of the weight dashpot is proved by the wiredrawing shown on the indicator diagrams of the Wheelock engine. The Wheelock engine was fitted with a Bulkley condenser, which does not require the use of an air-pump, the air present in the exhaust being carried down the descending limb of the condenser, by induction. The engines were tested, first, as condensing engines; and then as non-condensing engines.

Engine.	Diameter of Cylinder.	Stroke.	Total Clearance at each end.
Reynolds-Corliss	18.02 inches	4 feet	2.65 per cent.
Harris-Corliss	18.03 „	4 „	1.93 „
Wheelock	18.26 „	4 „	2.35 „

The cylinders were not steam-jacketed. The effective pressure in the boiler was 96 lbs. per square inch. The nominal speed of the engines was at the rate of 75 turns, or 600 feet of piston, per minute. Each engine was tested for a period of ten hours; the results are given in table No. 114. The steam from the boiler was tested for moisture by a "calorimeter," and the net steam delivered to the engine,—as dry steam,—is herein recorded. Sample indicator diagrams are given in figs. 234.

From the tabulated results on pages 516, 517, it appears that the ratios of the pressures are nearly the same as those of the respective actual expansions; and that the consumptions of steam per horse-power are as 5 to 6 for condensing and non-condensing action. Thus:

Averages.	Condensing.	Non-condensing.
Actual ratio of expansion	6.95	5.81
Ratio of pressures, initial and final	6.74	5.62
Steam per indicator horse-power per hour	19.58 lbs.	23.79 lbs.

The proportion of steam brought into evidence by means of the indicator-diagram, averages 81.30 per cent for the condensing engines, and 88.82 per cent for the non-condensing engines. The shortcoming is due to the portion of steam condensed in the cylinders, which is not restored by re-evaporation, in the absence of steam-jackets or equivalent means. But the proportion of deficit, averaging 18.7 per cent for the condensing engines, is less for the non-condensing engines, and averages 11.18 per cent only:—clearly due to the more elevated minimum temperature of the cylinders and the shorter range of temperature, when non-condensing, than when condensing.

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[1] See *Report of the Expert on the Test-trials of Automatic Cut-off Steam Engines, at the first Millers' International Exhibition*, Cincinnati, June, 1880.

Table No. 114.—RESULTS OF TEST-TRIALS OF AUTOMATIC CUT-OFF STEAM ENGINES. J. W. HILL. 1880.

	ENGINES CONDENSING.			ENGINES NON-CONDENSING.		
1. NAME OF ENGINE	REYNOLDS-CORLISS.	HARRIS-CORLISS.	WHEELOCK.	REYNOLDS-CORLISS.	HARRIS-CORLISS.	WHEELOCK.
2. Average period of admission ...	12.41%	11.87%	13.08%	15.96%	13.63%	16.97%
3. Net volume of cylinder	7.084 cu. ft.	7.092 cu. ft.	7.276 cu. ft.	7.084 cu. ft.	7.092 cu. ft.	7.276 cu. ft.
4. Total volume cut off...........	1.064 ,,	.978 ,,	1.121 ,,	1.316 ,,	1.103 ,,	1.405 ,,
5. Total final volume	7.269 ,,	7.228 ,,	7.445 ,,	7.269 ,,	7.228 ,,	7.445 ,,
6. Actual ratio of expansion.......	6.832	7.391	6.641	5.566	6.553	5.299
7. Total maximum pressure in cylinder	105.79 lbs.	104.77 lbs.	102.79 lbs.	104.69 lbs.	104.22 lbs.	103.20 lbs.
8. Total initial pressure at cut-off................	101.03 ,,	101.67 ,,	92.40 ,,	99.46 ,,	100.61 ,,	91.57 ,,
9. Total final pressure	15.16 ,,	14.57 ,,	14.04 ,,	17.41 ,,	17.04 ,,	17.46 ,,
10. Ratio of pressures, Nos. 8 and 9	6.668	6.978	6.581	5.714	5.904	5.244
11. Wiredrawing, difference of Nos. 7 and 8...	4.5%	3%	10%	5%	3.5%	11.3%
12. Weight of sensible steam cut off	.2478 lb.	.2291 lb.	.2401 lb.	.3022 lb.	.2559 lb.	.2984 lb.
13. Weight of sensible steam expanded	.2842 ,,	.2725 ,,	.2524 ,,	.3235 ,,	.3151 ,,	.3328 ,,
14. Difference of Nos. 12 and 13	.0364 ,,	.0434 ,,	.0123 ,,	.0213 ,,	.0592 ,,	.0344 ,,
15. Difference per cent of No. 13	12.81%	15.93%	4.87%	6.58%	18.78%	10.34%
16. Total work by expansion in heat units.......	36.48 units.	34.00 units.	36.36 units.	38.98 units.	40.74 units.	38.40 units.
17. Of which is provided by freed constituent heat	9.95 ,,	9.75 ,,	8.66 ,,	10.54 ,,	10.46 ,,	10.92 ,,
18. And by condensation of expanding steam	26.53 ,,	24.25 ,,	27.70 ,,	28.41 ,,	30.28 ,,	27.48 ,,
19. Weight of steam thus condensed	.0275 lb.	.0251 lb.	.0287 lb.	.0296 lb.	.0316 lb.	.0287 lb.
20. To be added to sensible expanded steam, No. 13	.2842 ,,	.2725 ,,	.2524 ,,	.3235 ,,	.3151 ,,	.3328 ,,
21. Total steam in evidence by indicator per stroke: sum of Nos. 19 and 20	.3117 ,,	.2976 ,,	.2811 ,,	.3531 ,,	.3467 ,,	.3615 ,,
22. Deduct compressed steam ..	.0077 ,,	.0063 ,,	.0071 ,,	.0234 ,,	.0329 ,,	.0315 ,,
23. Net steam in evidence consumed per stroke.......	.3040 ,,	.2913 ,,	.2740 ,,	.3297 ,,	.3138 ,,	.3300 ,,
24. Effective mean pressure per sq. inch	35.40 lbs.	35.67 lbs.	33.94 lbs.	29.78 lbs.	28.94 lbs.	29.36 lbs.
25. Turns per minute	75.38 turns.	75.83 turns.	74.47 turns.	75.33 turns.	75.81 turns.	76.07 turns.

Table No. 114 (*continued*).

	ENGINES CONDENSING.			ENGINES NON-CONDENSING.		
NAME OF ENGINE.	REYNOLDS-CORLISS.	HARRIS-CORLISS.	WHEELOCK.	REYNOLDS-CORLISS.	HARRIS-CORLISS.	WHEELOCK.
26. Speed of piston in feet per min.	603.04 feet.	603.44 feet.	595.76 feet.	602.64 feet.	606.48 feet.	608.56 feet.
27. Indicator horse-power............	165.0 H.P.	167.4 H.P.	160.4 H.P.	138.7 H.P.	135.8 H.P.	141.8 H.P.
28. Net steam in evidence consumed per hour, in cylinders.....	2750 lbs.	2651 lbs.	2449 lbs.	2980 lbs.	2855 lbs.	2949 lbs.
29. Net steam in evidence consumed per horse-power............	16.67 ,,	15.83 ,,	15.26 ,,	21.49 ,,	21.02 ,,	20.80 ,,
30. Steam delivered to the engine per hour.........	3360.6 ,,	3206 ,,	3085 ,,	3199 ,,	3216 ,,	3489 ,,
31. Steam delivered per horse-power	20.37 ,,	19.15 ,,	19.22 ,,	23.07 ,,	23.68 ,,	24.61 ,,
32. Difference of Nos. 28 and 30	610.6 ,,	555 ,,	636 ,,	219 ,,	361 ,,	540 ,,
33. Difference of per cent of No. 30	18.18°/₀	17.32°/₀	20.61°/₀	6.84°/₀	11.23°/₀	15.48°/₀
34. Net steam in evidence, per cent of steam delivered to the engine..........	81.82°/₀	82.68°/₀	79.39°/₀	93.16°/₀	88.77°/₀	84.52°/₀

Single-cylinder Condensing Engine, Steam-jacketed and Covered.—The Corliss cylinders, constructed by Messrs. Hick, Hargreaves & Co.,[1] and erected at Saltaire, have single cylinders 50 inches in diameter, with a stroke of 7 feet; thoroughly steam-jacketed with steam from the boiler, of 45 lbs. pressure. They make 30 turns per minute, consuming 2½ lbs. of coal, or (2½ lbs. x 6.96 =) 17.4 lbs. of water per indicator horse-power.[2] Four samples of indicator diagrams are given, figs. 235, for which the clearance-space is 2½ per cent. They respectively,

	1.	2.	3.	4.
Cut off at................................	25,	20,	15,	13 per cent.
Actual ratios of expansion............	3.36,	4.10,	5.30,	6.00
Total initial pressures for expansion	36½ lbs.,	37 lbs.,	32¾ lbs.,	32 lbs.
Total final pressures, taken at 90 per cent of the stroke............	11 ,,	8¾ ,,	6½ ,,	6½ ,,
Ratios of pressures.....................	3.27,	4.23,	5.04,	5.00

These ratios nearly agree with those of the expansions, and the light-line hyperbolic curves on the diagrams show a general approximation of the actual curves of expansion to the hyperbolic curves. The weights of steam cut off, and expanded when the piston reaches 90 per cent of the stroke, are:—

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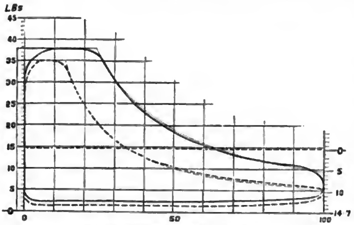
[1] See Mr. W. Inglis' paper on Corliss valve-gear, in the *Proceedings of the Institution of Mechanical Engineers*, 1868; page 177.

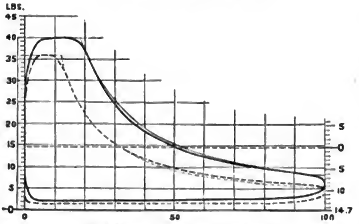
[2] The multiplier, 6.96, is given by Mr. Hopkinson.

Cut off……………………………	2.34 lbs.,	2.13 lbs.,	1.34 lbs.,	1.14 lbs.
Expanded……………………………	2.48 ,,	2.00 ,,	1.54 ,,	1.54 ,,
Excess weight expanded……………	6°/。	6°/。	15°/。	35°/。

The average effective mean pressures for the four diagrams is 16.74 lbs. per square inch, and (16.74 lbs. × 1963.5 square inches of piston × 7 feet stroke × 2 × 30 turns per minute) ÷ 33,000 = 418.3 indicator horse-power for one cylinder. The net steam consumed per hour, deducting the small fractions detained by compression, averaged from the four diagrams, is 6070 pounds per hour, as cut-off, or 6705 pounds per hour, as expanded; or 14.51 pounds and 16.03 pounds per indicator horse-power. The actual quantity consumed has already been calculated as 17.4 pounds per horse-power. The excess of this over the initial quantity, 14.51 pounds, is 2.89 pounds, which is 20 per cent over, and may be taken as the quantity of steam condensed during admission to the cylinder.



Cut off at 25 per cent and 15 per cent.



Cut off at 20 per cent and 13 per cent.

Figs. 235.—Corliss Cylinders, Saltaire.

Mr. Hopkinson[1] records the results of trials conducted by Mr. George Salt, on the comparative results of performance when steam was in the jackets, and when it was shut off, in one pair of Corliss cylinders. The quantities of coal and of water consumed were respectively as 6 to 7, in the two cases, showing an economy of one-seventh or 14.3 per cent, by employing the steam-jacket. The steam used in the jackets amounted to one-thirtieth, or 3.33 per cent of the total used.

A *single-cylinder condensing beam engine* of 100 horse-power, at the factory of Messrs. Haussmann, Jordan, Hirn & Co., Logelbach, has been the subject of many tests with the use of superheated steam. The cylinder is 23¾ inches in diameter, with a stroke of 67 inches. It is not steam-jacketed. It is lagged with 2-inch oak in strips, inclosing an interspace 1⅛ inches wide. It is fitted with four valves—two for admission, and two for exhaust—and short passages, making a total clearance measured by 1 per cent of the stroke, at each end of the cylinder, for admission.

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[1] *The Indicator and the Steam Engine*, 1875; page 213.

A course of eight trials of this engine, during eight days, in 1873 and 1875, was conducted by M. G. A. Hirn, and made by MM. Dwelshauvers-Dery, W. Grosseteste, and O. Hallauer.[1] The steam was generated at 50 lbs. effective pressure, and was, for the purpose of the trials, not only used superheated, but also used in the ordinary saturated condition. The conditions of the first and second trials differ from the others in this, that, after those were made, and before these were made, the cylinders were renewed, with enlarged exhaust-ports and exhaust lead, the effect of which was a comparative economy of 9 per cent. The leading results of these trials are given in the table No. 115, which has, for the most part, been calculated on the same lines as previous tables. The actual ratio of expansion, line 5, is calculated to the end of the stroke. In the calculation of the weight of sensible steam cut off, line 10, the steam is, in each instance, taken as saturated, for though it has been superheated in several instances, the state of saturation at the point of cut-off is proved by the major weights of steam consumed from the boiler, line 20, except only in one instance, No. 4, in which the indicated weight of steam cut off, taken as saturated, was slightly in excess of the weight of steam consumed from the boiler. In this case, there is evidence of a slight degree of superheat at the point of cut-off. The weight of the expanded steam, line 11, has also been reckoned for a state of saturation. The calculation of work by expansion, lines 14, 15, 16, and 17, has been made in the same manner as in previous cases already explained. In selecting the total weights of steam per stroke, in evidence by the indicator, lines 18 and 19, the maximum weight in each instance is selected and entered in the table, in line 18 or line 19, according as the initial weight, line 10, or the weight of the expanded steam, line 11, plus the weight of steam condensed in expansion, was the greater. The whole of the water consumed from the boiler, line 20, is reckoned as steam, as it was ascertained that priming, if it did exist, was but trifling. No deduction has been made by the experimentalists for exhaust-steam compressed and retained in the cylinder, but it is said to be trifling in proportion to the total quantity of steam consumed. There are no means given in the report for calculating the weight of steam thus retained, except for trial No. 8, when steam of 15 lbs. absolute pressure was exhausted into the atmosphere, and was shut in for compression at 7½ per cent of the stroke. It has been calculated to be .0501 pound in weight, for one stroke, and in order to approach as nearly as may be to accuracy, this weight has been deducted from the calculated gross weight, delivered into the cylinder, leaving the balance consumed, .6544 lb., as entered for No. 8.

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[1] *Bulletin de la Société Industrielle de Mulhouse*, 1877; page 144

Table No. 115.—M. Hirn's Condensing Engine: Superheated Steam. Performance of the Engine.

Steam in boilers, 65 lbs. average absolute pressure; 298° F. temperature.

1. No. of trial		1.	2.	3.	4.	5.	6.	7.	8.
2. Condition of steam		super-heated	satu-rated	super-heated	super-heated	super-heated	satu-rated	super-heated	sup'heat'd (not condensed)
3. Degrees of superheat	Fah.	150°	—	121°	135°	85°	—	130°	130°
4. Period of admission	%	25.7	25.7	21.4	45.4	16.3	16.3	45.4	28.7
5. Actual ratio of expansion	ratio	3.75	3.75	4.51	2.18	5.84	5.84	2.18	3.40
6. Total maximum pressure in cylinder	lbs.	69.35	59.00	63.28	64.79	66.64	61.51	37.32	59.81
7. Total initial pressure for expansion	,,	60.35	53.70	58.88	32.79	55.64	54.51	24.82	48.81
8. Total final pressure	,,	13.30	13.60	10.98	11.96	8.43	8.16	9.18	15.71
9. Ratio of pressures	ratio	4.54	3.95	5.36	2.74	6.60	6.68	2.70	3.11
10. Weight of sensible steam cut off	lb.	.6318	.5670	.5817	.6320	.3722	.3656	.4874	.5788
11. Do. do. expanded	,,	.5949	.6156	.4820	.5402	.3883	.3764	.4203	.6575
12. Difference of Nos. 10 and 11	,,	−.0369	+.0486	−.0997	−.0918	+.0161	+.0108	−.0671	+.0787
13. Do. per cent of No. 11	%	−6.2	+7.89	−20.68	−17.00	+4.15	+2.88	−15.97	+11.97
14. Total work by expansion	heat units	66.08	63.32	63.16	36.96	57.24	54.80	28.88	58.00
15. Of which there is constituent heat of steam freed by expansion	,,	16.50	14.47	16.69	10.37	12.11	11.82	7.55	12.82
16. And heat of steam condensed for work by expansion	,,	49.58	48.85	46.47	26.59	45.13	42.98	21.33	45.18
17. Weight of steam so condensed	lb.	.0511	.0505	.0476	.0273	.0459	.0438	.0217	.0470
18. Total weight of steam per stroke, in evidence by indicator, sum of Nos. 11 and 17	,,	.6460	.6661	.5296	.5475	.4342	.4202	.4420	.6544
19. Do. do. as in No. 10.	,,	—	—	.5817	.6320	—	—	.4874	—
20. Water as steam from boiler consumed per stroke	,,	.6758	.8229	.5845	.6222	.4939	.5808	.4994	.6575
21. Difference of Nos. (18 and 19) and 20	,,	+.0298	+.1568	+.0028	−.0108	+.0597	+.1606	+.0120	+.0031
22. Do. per cent of No. 20	%	+4.4	+19.1	+.50	−1.7	+12.1	+27.6	+2.4	+.47
23. Effective mean pressure	lbs.	26.71	24.94	25.29	23.32	21.06	19.79	18.44	14.77
24. Turns per minute	turns	30.17	30.55	29.97	30.31	29.98	30.41	30.13	30.00
25. Speed of piston per minute	feet	336.7	340.9	334.5	338.3	334.6	339.4	336.3	334.8
26. Indicator horse-power		142.4	134.6	133.9	123.5	111.6	106.3	98.1	77.2
27. Calculated water as steam (Nos. 18 & 19) consumed per H.P. per hour	lbs.	16.42	18.14	15.62	18.61	14.20	14.42	17.96	30.50
28. Water as steam actually consumed from the boiler per H.P. per hour	,,	17.18	22.41	15.69	18.33	16.16	19.93	18.40	30.65
29. Difference of Nos. 27 & 28	p.cent	+4.4	+19.1	+.50	−1.7	+12.1	+27.6	+2.4	+.47

Correcting, for comparison, Nos. 1 and 2, by 9 per cent, for the reason already given, the corrected actual quantities of water consumed per horse-power per hour, line 28, are 15.63 lbs. and 20.39 lbs. The comparative results of greatest efficiency are, then, as follows:—

	Fahr.	Cut-off.	Ratio.	Water per Horse-power per Hour.
No. 1 (corrected), superheated	150°	25.7 %	3.75	15.63 lbs.
No. 3 do.	121	21.4	4.51	15.69 „
No. 5 do.	85	16.3	5.84	16.16 „
No. 2 (corrected), saturated		25.7	3.75	20.39 „
No. 6 do.		16.3	5.84	19.93 „

It appears that the economy effected by superheating the steam to the extent of 150° F., comparing Nos. 1 and 2, is 23 per cent, and that in superheating 85°, comparing Nos. 5 and 6, it is 20 per cent. The greater economy is effected with the greater degree of superheat conjointly with the lower ratio of expansion. The evidence points to the conclusion that the expansion-ratio 4.5, when the steam is cut off at about one-fifth, is conducive to the maximum of economy, for superheated steam. For saturated steam all that can be inferred is that, in round numbers, the ratios 4 and 6 are nearly equal in economical results, and that some intermediate ratio is the most effective for economy; also that from 4 pounds to 5 pounds more of saturated than of superheated steam is consumed per indicator horse-power per hour.

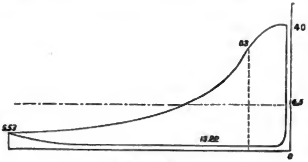
The comparative gain by the use of the condenser in the use of superheated steam, as against the exhaustion of the steam into the atmosphere, in No. 8 trial, is represented by the steam consumed per horse-power in No. 1 trial corrected, which is the nearest example for comparison. For No. 1, 15.63 pounds of steam are consumed per horse-power, being only one-half of the steam, 30.65 lbs., consumed, without condensation, in No. 8 trial. The difference is deducible in another way, on considering that as the effective mean pressure, in No. 8, is 14.77 lbs. (line 23), it would be just doubled by the addition of one atmosphere of pressure, or 14.7 lbs. per square inch.

The steam actually consumed has been fairly well brought into evidence by the indicator, in the cases of superheated steam, as shown by the small percentages of deficiency in line 22; except in the instance of No. 5, cutting off at 16.3 per cent, or one-sixth, having the highest ratio of expansion, 5.84, in which the indicator fails by 12.1 per cent to account for the whole of the steam. In this instance, the superheat has been carried to the minimum extent, whereas for a minimum admission and relatively great expansion the superheat should have been relatively a maximum in order to prevent internal condensation. Accordingly, in the instances of saturated steam without superheat, No. 2 and No. 6, the differences or deficiencies of steam brought into evidence by the indicator, line 22, amounted to 19.1 per cent and 27.6 per cent respectively, cutting off at one-fourth and one-sixth.

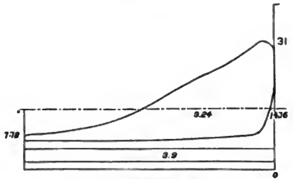
Here it is apparent that, generally, as the extent of primary superheating is reduced, the deficiency, or steam not accounted for through the indicator, is augmented; and when saturated steam is used, it rises to a maximum. The existence of such a relation is inexplicable, except on the supposition that there was considerable loss of heat by external radiation or conduction from the cylinder, occasioned by insufficient covering in combination with a comparatively limited speed of piston, of about 336 feet per minute for 30 revolutions per minute.

The ratios of the pressures, initial and final, of expansion are, in the instances of superheated steam, excepting No. 8, materially greater than the expansion-ratios. That is, the pressures fall more rapidly than the steam expands, or in a ratio more nearly adiabatic than hyperbolic. This comparatively rapid fall of pressure is in harmony with the behaviour of superheated steam in expansion already noticed, page 495. In the case of No. 8, in which the steam is exhausted into the atmosphere, the average temperature of the cylinder must be higher than in the cases in which the steam is condensed, and possibly the comparatively less rapid fall of pressure in expansion, for No. 8, is due to that fact.

Single-cylinder Beam Engines, Condensing.—Mr. John G. Mair has made a number of exact tests for efficiency, the results of which are given in table No. 116. Two of these results, A and K, are derived from Mr. Mair's unique paper on "The Independent Testing of Steam Engines."[1] The others are described in Mr. Mair's second paper. The indicator diagrams, figs. 236, are illustrative of the tests A and K. They are the averages of each series of diagrams taken for the tests. Trial A was made with a single-cylinder beam rotative engine, made by Simpson & Co., working a deep well and a high-lift pump. The cylinder is 45 inches in diameter, with a stroke of 5½ feet, and is thoroughly steam-jacketed. The steam is distributed by separate slides at the top and bottom of the cylinder with adjustable expansion-plates, worked by eccentrics. Three trials were made with the engine, giving the same result.



Engine A.



Cornish Engine, K.

Figs. 236.—Mr. Mair's Test Trials.

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[1] "The Independent Testing of Steam Engines, and the Measurement of Heat used," and "The Results of some Independent Engine-tests," in the *Minutes of Proceedings of the Institution of Civil Engineers*, vol. lxx. 1881–2, page 313; vol. lxxix. 1884–5, page 323.

Trials L, M, were made with a single-cylinder beam rotative engine, having a 32-inch cylinder with a 5½-feet stroke. The walls and base of the cylinder are steam-jacketed direct from the boiler, and the distribution is effected by main and expansion slides at the top and the bottom, worked by excentrics. The cut-off is adjustable by hand. Trials N, O, were also made with a single-cylinder beam rotative engine, similar to the engines for the trials L, M, and supplied with steam from the same group of boilers; having a 27-inch cylinder with a 6-feet stroke.

Trial K was made with a single-acting beam Cornish engine, working a single-acting piston-pump, constructed by Harvey & Co. The cylinder is 68¼ inches in diameter, with a stroke of 8 feet, and is steam-jacketed at the sides only. It has the usual steam, equilibrium, and exhaust valves. The total clearance space has been estimated at 3½ per cent for the purpose of calculating the expansion ratio.

The results of the tests of the above-noted engines, in table No. 116, are arranged in the order of the expansion-ratios. In trial O, the expansion-plates were set to cut off at half-stroke, and the steam had therefore to be wiredrawn.

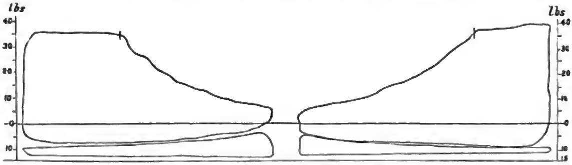
Table No. 116.—Results of Tests of Single-cylinder Steam Engines.
Mr. J. G. Mair. 1883.

Engine, and No. of Test.	Cylinder.		Steam-jacketing.	Net Pressure in Boiler per square inch.	Period of Admission.	Actual Ratio of Expansion.	Total Maximum Pressure in Cylinder per sq. in.	Total Initial Pressure at Cut-off per square inch.	Turns per Minute.	Speed of Piston in Feet per Minute.	Indicator Horse-power.	Dry Saturated Steam per Horse-power per Hour.	Water as Steam from the Boiler per Horse-power per Hour.
	Diameter.	Stroke.											
1	2	3	4	5	6	7	8	9	10	11	12	13	14
	inches.	inches.		lbs.	p. cent.	ratio.	lbs.	lbs.	turns.	feet.	H.P.	pounds.	pounds.
A	45	66	Thorough.	33	13.2	6.85	40	33	14.62	160.8	101.79	22.06	22.94
L	32	66	Vertical case.	42	21.8	4.33	55.5	45.93	20.31	223.4	123.36	22.06	22.28
M	32	66	Steam shut off.	41.1	24.6	3.84	54.5	45.68	20.26	222.9	123.6	26.46	26.72
N	27	72	Vertical case.	44.7	30.0	3.16	55.45	44.45	20.00	240	119.5	21.3	21.51
O	27	72	Vertical case.	44.6	50.0	1.95	53.6	33.6	20.00	240	122.4	23.6	23.84
K (Cornish)	68¼	96	Vertical case.	31.5	50.0	1.93	31	16	12.8 (strokes)	201.3	146.0	24.15	24.87

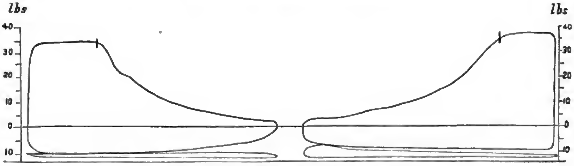
The proportion of the diameter and the stroke to each other, appears to influence the efficiency of these engines to some extent. Engines L and N, both jacketed at the sides only, contrast in this respect, N having the longer stroke proportionally, consuming less steam than L. The advantage of thorough jacketing against partial jacketing is shown by A engine, which, though less favourably proportioned, with lower pressures, and cutting off too early for the greatest efficiency, consumes just the same weight of steam as engine L. The advantage of the steam jacket against the steamless jacket is shown by L and M tests of the same engine, in which 20 per cent more steam was used when there was no steam in the jacket.

CHAPTER IX.—PRACTICE OF EXPANSIVE WORKING IN COMPOUND-CYLINDER STEAM ENGINES.

Compound Woolf Beam Engine, Condensing.—The factory engine of Bryan Donkin & Co., London, is a Woolf engine, of which the first cylinder is 13⅜ inches in diameter, with a stroke of 3 feet 3⅜ inches; and the second cylinder is 24 inches by 4½ feet. The net capacities of the cylinders are 3.116 cubic feet, and 14.137 cubic feet respectively; and their ratio is 4.537. The total clearance of the first is 10 per cent, and that of the second is 3.77 per cent at the top, and 4.8 per cent at the bottom. The intermediate space between the cylinders, from the exhaust of the first to the piston of the second, is 1.463 cubic feet, or 10.33 per cent of the capacity of the second cylinder at the top and 1.609 cubic feet, or 11.36 per cent at the bottom. Each cylinder is worked with a slide-valve, cutting off at seven-eighths of the stroke; and the first cylinder is fitted with an expansion valve in addition, to cut off at any earlier point of the stroke. Both cylinders are steam-jacketed at the sides and the bottoms; but not at the upper ends. The steam from the boiler for the working of the engine is conducted directly to the bottoms of the cylinders, from which it passes upwards within the jacket inclosing the two cylinders, to the upper part, whence it passes to the valve-chest of the first cylinder.



Figs. 237.—Indicator Diagrams · Steam in Jackets.



Figs. 238.—Indicator Diagrams: Steam not in Jackets.
B. Donkin & Co.'s Factory Woolf Engine.

During a 10-hours trial of this engine, with steam in the jackets, made by W. B. Farey and Bryan Donkin, junr., the pressure in the boiler was 40 lbs. per square inch. The indicator diagrams, figs. 237, were taken from the cylinders in the course of the trial, when the engine made 32.48 turns, or 213 feet of the first piston per minute. The steam was cut off in the first cylinder, at 38 per cent at the top, and 31 per cent at the bottom. The exhaust-steam of the first cylinder was shut in for compression during 5 per cent of the return-stroke at the top, and 10 per cent at the bottom. The actual ratios of expansion in the first cylinder were 2.291 for the top, and 2.682 for the bottom.

The same Woolf engine was tried when the steam was shut off from the jackets. The indicator diagrams, figs. 238, were taken during the trial, showing that the steam was cut off at 28 per cent at the top, and 22 per cent at the bottom. The actual ratios of expansion in the first cylinder were 2.893 for the top, and 3.436 for the bottom.

The intermediate expansion-ratios, supposing the intermediate space to be vacuous, would have been 1.470 for the top of the first cylinder, and 1.427 for the bottom. But, as before stated, these ratios should be corrected in consideration of the residual steam left in the intermediate space, after it is cut off from the second cylinder. For this and other determinations, the following absolute pressures of steam are given. The final pressures are taken as for the ends of the stroke, and are determined by the tensions of the expansion-curves:—

Absolute Pressures of Steam.	I. Steam in Jackets.		II. Steam not in Jackets.	
	top.	bottom.	top.	bottom.
Cut off in 1st cylinder	50.3 lbs.	51 lbs.	47 lbs.	50.2 lbs.
Expanded in do.	21 „	19.75 „	17 „	17.2 „
In 1st cylinder when exhausted into intermediate space	13.5 „	12 „	9.75 „	8 „
	bottom.	top.	bottom.	top.
Initial in 2d cylinder	11 „	11 „	4.3 „	5.4 „
Terminal in do.	4.7 „	5.2 „	3.5 „	3.2 „
	top.	bottom.	top.	bottom.
For compression in 1st cylinder	8 „	7 „	4.2 „	6 „
Do. do. in 2d do.	2 „	2.7 „	3 „	2.2 „

When steam is in the jackets, the steam in the first cylinder, at the top, is expanded from 13½ lbs. to 4.7 lbs. when cut off from the second cylinder; and, conversely, it is compressed from 4.7 lbs. to 13½ lbs. by the succeeding discharge of steam from the first cylinder into the intermediate space. This space, 1.609 cubic feet in volume, is occupied by that remnant of steam, when thus compressed, to the extent of $\left(1.609 \times \frac{4.7}{13.5} = \right)$ .555 cubic foot, and the difference, $(1.609 - .555 =)$ 1.054 cubic feet, is the volume of the intermediate space actually occupied by the newly exhausted steam from the first cylinder, and is the measure of the intermediate expansion. The volume of the first cylinder plus clearance at the top, is 3.426 cubic feet;

and the actual intermediate expansion, from 3.426 to (3.426+1.054=) 4.480 cubic feet, is only in the ratio 1 to 1.308, and not 1 to 1.470, the nominal ratio. The volume, 4.480 cubic feet, is the initial volume for expansion from the bottom of the second cylinder. Similarly, for the bottom of the first cylinder, the intermediate steam expanded from 12 lbs. to 5.2 lbs., is compressed, conversely, from 5.2 lbs. to 12 lbs., occupying $\left(1.465 \times \frac{5.2}{12} = \right)$ .761 cubic foot, and leaving (1.463−.761=) .702 cubic foot of intermediate space for fresh exhaust-steam from the first cylinder. Adding this to the gross volume of the first cylinder, 3.426 cubic feet, the ratio of this volume to the intermediately expanded volume, (3.426 +.702=) 4.128, is as 1 to 1.205, instead of 1.427 the nominal ratio. This volume, 3.426 cubic feet, is the initial volume for expansion for the top of the second cylinder.

Again, the final expanded volume in the second cylinder, including the clearance of the first cylinder, is (14.137+1.919=) 16.056 cubic feet for the bottom; and (14.137+1.773=) 15.910 cubic feet for the top; and the actual expansion-ratios are as 4.480 to 16.056, or as 1 to 3.584 for the bottom; and as 4.128 to 15.910, or as 1 to 3.854 for the top of the second cylinder.

Adjusting the intermediate ratios similarly for the steamless jackets, the volume of intermediate space for the steam discharged from the first cylinder is 1.032 cubic feet for the top, and .965 cubic foot for the bottom of the first cylinder. These volumes, added to the volume of the first cylinder, 3.426 cubic feet, make the initial volumes for expansion into the second cylinder, 4.458 cubic feet at the bottom, and 4.391 cubic feet at the top. The ratios of intermediate expansion are, correspondingly as 3.426 to 4.458, or as 1 to 1.301; and as 3.426 to 4.391, or as 1 to 1.282.

The actual ratios of expansion into the second cylinder are as 4.458 to 16.056, or as 1 to 3.602 for the bottom; and as 4.391 to 15.910, or as 1 to 3.623 for the top.

The expansion-ratios are brought together, thus:—

	I. Steam in Jackets.		II. Steam not in Jackets.	
Actual ratio of expansion (1) 1st cylinder ..	top 2.291	bot'm 2.682	top 2.893	bot'm 3.436
Do. do. (2) intermediate.	,, 1.308	,, 1.205	,, 1.301	,, 1.282
Do. do. (3) 2d cylinder ..	bot'm 3.584	top 3.854	bot'm 3.602	top 3.623
Total expansion-ratio	10.735	12.450	13.556	15.912
Net do. (1) and (3).....	8.211	10.340	10.420	12.412
Mean total expansion-ratio	11.592		14.734	
Mean net or working do.	9.276		11.416	
Ratio of initial and final pressures by expansion:—				
(1) 1st cylinder........................	top 2.400	bot'm 2.590	top 2.765	bot'm 2.920
(2) intermediate........................	,, 1.910	,, 1.791	,, 3.953	,, 3.185
(3) 2d cylinder....	bot'm 2.340	top 2.116	bot'm 1.230	top 1.690
Total ratio	10.730	9.815	13.440	15.720
Net or working ratio (1) and (3)......	5.616	5.480	3.401	4.935
Mean total ratio.........................	10.273		14.580	
Mean working ratio	5.548		4.168	

The disposal of the steam under the two conditions of steam-jacketing and non-steam-jacketing, is developed at length in the following statement:—

	I. Steam in Jackets.		II. Steam not in Jackets.	
1. Sensible steam cut off, 1st cylinder....	top .1808 lb.	bot'm .1563 lb.	top .1342 lb.	bot'm .1176 lb.
2. Do. do. expanded, do. do.	,, .1819	,, .1716	,, .1490	,, .1507
Difference.....................	.0011	.0153	.0148	.0331
Do. per cent of No. 2	0.60°/₀	8.90°/₀	10°/₀	20.91°/₀
Mean..................	4.75°/₀		15.46°/₀	
3. Sensible initial steam for expansion into 2d cylinder..	bot'm .1295 lb.	top .1193 lb.	bot'm .0539 lb.	top .0650 lb.
4. Sensible steam expanded into 2d cylinder	,, .2087	,, .2275	,, .1573	,, .1432
Difference of Nos. 3 and 4..	.0792	.1082	.1034	.0782
Do. per cent of No. 4	37.95°/₀	47.56°/₀	65.73°/₀	54.61°/₀
Mean..................	42.76°/₀		60.17°/₀	
Difference of Nos. 1 and 4..	.0279 lb.	.0712 lb.	.0231 lb.	.0256 lb.
Do. per cent of No. 4	13.37°/₀	31.30°/₀	14.68°/₀	17.90°/₀
Mean..................	22.33°/₀		16.29°/₀	
	1st Cylinder.	2d Cylinder.	1st Cylinder.	2d Cylinder.
5. Mean units of work-heat per stroke by expansion.....	15.07 units	18.72 units	13.86 units	8.18 units
6. Of which are provided by freed constituent heat........	2.83 ,,	2.36 ,,	2.71 ,,	.74 ,,
7. And by condensation of expanding steam	12.24 ,,	10.36 ,,	11.15 ,,	7.44 ,,
8. Weight of steam thus condensed	.0129 lb.	.0164 lb.	.0116 lb.	.0073 lb.
9. To be added to the mean sensible expanded steam	.1768 ,,	.2181 ,,	.1499 ,,	.1502 ,,
10. Total steam in evidence—sum of Nos. 8 and 9, the weights of condensed steam in both cylinders being added for the 2d cylinder............	.1897 ,,	.2474 ,,	.1615 ,,	.1691 ,,
11. Deduct compressed steam ...	.0109 ,,	.0046 ,,	.0068 ,,	.0050 ,,
12. Net steam, in evidence, consumed per stroke	.1788 ,,	.2428 ,,	.1547 ,,	.1641 ,,
13. Effective mean pressure per square inch.....................	30.35 lbs.	4.76 lbs.	26.35 lbs.	1.465 lbs.
14. Turns per minute.............	32.48	32.48	32.02	32.02
15. Speed of piston, in feet per minute........	213 feet	292 feet	210 feet	288 feet
16. Indicator horse-power	27.53 H.P.	19.09 H.P.	23.56 H.P.	5.79 H.P.
17. Combined do. do.	46.62 H.P.		29.35 H.P.	
18. Net steam, in evidence, consumed per hour in cylinders	697 lbs.	946 lbs.	594.4 lbs.	630 lbs.
19. Do. do. do. per I.H.P...	14.95 ,,	20.30 ,,	21.40 ,,	21.48 ,,
20. Average indicator horse-power for the whole period of the trial, lasting 10 hours, speed as above stated........	46.21 H.P.		27.78 H.P.	
21. Water as steam consumed from the boiler, per hour....	1040.4 lbs.		909.4 lbs.	
22. Do. do. do., per H.P....	22.32 ,,		32.72 ,,	
23. Of which there was condensed in the jacket per H.P.	3.82 ,,		—	
24. And consumed within the cylinders per H.P.......... ..	18.50 ,,		32.72 lbs.	

The inequalities of the periods of admission for the top and bottom of the first cylinder, and as for steam-jacketing and non-jacketing, complicate the work of comparison. But the influence of the steam-jacket is obviously very considerable in minimizing the alternate condensation and re-evaporation of steam in the first cylinder, the mean difference of the initial and final weights of expanded steam, when jacketed, being 4.75 per cent, against 15.46 per cent not jacketed. The excess difference in the second case arises, no doubt, in part, from the shorter periods of admission, a mean of 25 per cent against 35 per cent when jacketed. The same preponderance prevails in the second cylinder, wherein the mean difference of weights of sensible steam, initial and final, is 42.76 per cent of the final weight, for steam-jacketing, and 60.17 per cent without steam-jacketing. These unusually large percentages, indicative of much condensation and re-evaporation, are due partly to the vastly increased cooling surface of cylinders, compared with the first cylinder alone, presented by the fact of the second expansion taking place in two cylinders plus the intermediate space; partly to the special proportions of the cylinders, of which the strokes are three times and more than twice the diameters respectively—by which the cooling surface is still further augmented, as compared with cylinders of more ordinary proportions; and partly to the very low speeds of the pistons. These circumstances, at the same time, serve to explain the unusual economy effected by the employment of the jacket, which has the effect of reducing the gross consumption of steam from the boiler by one-third, or from 32.72 lbs. to 22.32 lbs. per indicator horse-power per hour (line 22). But this has not been done without a large consumption of steam in the jacket, being 3.82 lbs. per indicator horse-power per hour (line 23), or 20.5 per cent of the steam consumed in the cylinders, by which the consumption is augmented from 18.50 lbs., used in the cylinders, to 22.32 lbs.

The action of the jacket in saving the steam from extinction in the second cylinder is made apparent by the comparative initial weights, line 3, which, though very much less in both cases than the final sensible weights in the first cylinder, are less reduced with steam in the jacket than without it. They average 70 per cent and 40 per cent respectively of the final sensible weights in the first cylinder.

This sudden fall is but an episode in the course of the expansion, for, practically, the expansion that takes place in the second cylinder is continuous with that in the first cylinder, the two cylinders being in direct communication with each other during the expansion in the second cylinder. The expansion in the two cylinders taken together is practically one operation. The range and fall of temperature in the first cylinder is, of course, greater than it would be if it were entirely separated from the second cylinder; and the resuscitation of steam that takes place during expansion in the second cylinder is supported in part by the reconversion into steam of condensation-water in the first cylinder.

The net steam brought into evidence by the indicator diagram, as consumed, is 20.30 lbs. per horse-power per hour, with steam in the jacket (line 19). This is exclusive of the steam used in the jacket, and it is in excess of the quantity of steam used from the boiler, averaged for the whole period of the trial, which is 18.50 lbs. per horse-power per hour. Such apparent excess is that of a special result over an average result, and may have arisen from the diagram not having happened to be an exactly average diagram.

The pressures by expansion in the first cylinder fall approximately in the inverse ratios of the expansions; but in the second cylinder there is no approximation to the hyperbolic ratio of the pressures.

A special series of comparative trials of the factory engine was made, for the purpose of determining the relative economy of different ratios of expansion:—

Ratio of expansion[1]...............	18	9	4
Pressure in the boiler	40 lbs.	40 lbs.	15 lbs.
Indicator horse-power.............	39.83	49.12	39.04
Water consumed from the boiler per horse-power per hour......	24.07 lbs.	23.89 lbs.	27.61 lbs.

There is nothing here to show that one ratio of expansion is better than another. The only deduction that can be made is that an expansion-ratio of 9 is at least as good as any other.

In another series of trials for the same object, with a Woolf engine of a smaller size, the rate of expansion was the same with steam and without steam in the jacket, namely 9, with 45 lbs. steam in the boiler. The quantities of water consumed were respectively 24¼ lbs. and 39½ lbs. per indicator horse-power.

Two other compound engines, constructed by the same firm, alike in all respects, were tested with 40-lb. steam, expanding four times: one with steam in the jacket, and indicating 140 horse-power, the other without steam in the jacket, and indicating 90 horse-power. With steam in the jacket, from 23 to 25 per cent less steam was consumed per horse-power.[2]

A *horizontal tandem compound condensing engine*, constructed by Messrs. Bryan Donkin & Co., and erected at the Hele Paper Works, Cullompton, Devon, was tested October 11, 1871.[3]

The cylinders, fig. 239, measured when hot, were $16^{17}/_{32}$ inches and $30^{3}/_{16}$ inches in diameter, with a stroke of 3 feet. The net capacities of the cylinders are 4.458 cubic feet, and 14.826 cubic feet, having the ratio 3.33. The total clearance of the first cylinder is .357 cubic foot, or 8 per cent of the net capacity; and the intermediate space, including the clearance of the second cylinder, is 1.705 cubic feet, or 11½ per cent of the net capacity. The first cylinder is placed nearest to the crank-shaft, the second cylinder being behind it, in line. They are worked with a slide-valve to each, the two valves being worked by one rod and excentric, and cutting off at seven-eighths, or 87.5 per cent of the stroke. A variable expansion-valve is placed on the back of the slide-valve of the first cylinder. The second cylinder only is jacketed—on the cylindrical surfaces only—the steam for the supply of the engine being led through the jacket, and delivered thence to the valve-chest of the first cylinder. The condensation-water of the jacket is drained off through a trap.

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[1] These ratios of expansion are stated just as they were given to the author; he cannot explain how they were calculated.

[2] *Engineering*, June 5, 1874; page 409.

[3] See *Engineering*, November 3, 1871, which contains a detailed report of the trial.

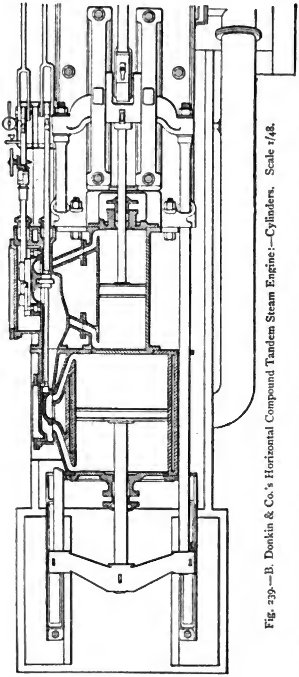


Fig. 239.—B. Donkin & Co.'s Horizontal Compound Tandem Steam Engine:—Cylinders. Scale 1/48.

The trial lasted for ten hours continuously. Sample indicator diagrams are given in figs. 240, page 532. The pressure in the boiler averaged 53 lbs. net per square inch; the speed averaged 46.51 turns per minute, or 279 feet of piston. The steam was cut off in the first cylinder at 12½ per cent for the top or crank end, and 14 per cent for the other end, making, with clearance included, the initial volumes for expansion, .9139 cubic foot, and .9808 cubic foot. The total pressure at the cut-off is 51.5 lbs. per square inch, at each end, and the final pressure, at the end of the stroke, is 10 lbs. at each end, from which it falls to 7 lbs. and 8½ lbs. initial pressures in the second cylinder, at the bottom and at the top, in which it expands to 4½ lbs. and 3¾ lbs. respectively, of final pressure reckoned for the ends of the strokes, the final volumes by expansion being 4.815 cubic feet. The actual ratio of expansion in the first cylinder is, for the top, 5.269; and for the bottom, 4.909. The intermediate space for expansion is reduced from 1.705 cubic feet to .938 cubic foot for the top, and to 1.066 cubic feet for the bottom, by residual steam, calculated in the manner already explained, page 525; and the actual ratio of expansion into the intermediate space is as 4.815 to (4.815 + .938 =) 5.753, or as 1 to 1.95 for the top; and as 4.815 to (4.815 + 1.066 =) 5.881, or as 1 to 1.221 for the bottom. The actual ratio of expansion into the second cylinder, reckoned to the end of the stroke, is, for each end, as 6.520 to 16.888 cubic feet (which includes the clearance-space of the first cylinder), or as 1 to 2.590.

The actual expansion-ratios, in the three consecutive stages, are brought together, as follows:—

Actual ratio of expansion (1) 1st cylinder	top	5.269	bottom	4.909	mean	5.089
Do do. (2) intermediate	,,	1.195	,,	1.221	,,	1.208
Do. do. (3) 2d cylinder	,,	2.590	top	2.590	,,	2.590
Total expansion-ratio		16.310		15.530		15.920
Net do. do. (1) and (3)		13.650		12.720		13.185

Ratio of initial and final pressures by expansion:—

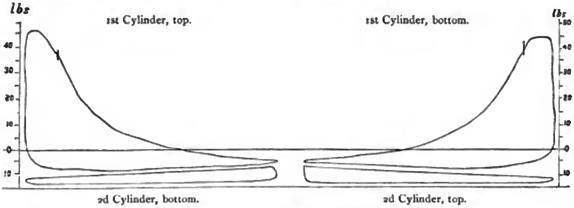
(1) 1st cylinder	top	5.150	bottom	5.150	mean	5.150
(2) intermediate	,,	1.430	,,	1.177	,,	1.304
(3) 2d cylinder	bottom	1.555	top	2.267	,,	1.911
Total ratio		11.450		13.740		12.830
Net or working ratio		8.008		11.680		9.842

The weights of sensible steam for different points of the system are as follows:—

1. Sensible steam cut off, 1st cylinder	top	.1129 lb.	bottom	.1211 lb.	mean	.1170 lb.
2. Do. expanded ,,	,,	.1271 ,,	,,	.1271 ,,	,,	.1271 ,,
Difference of Nos. 1 and 2		.0142		.0060		.0101
Do. per cent of No. 2		11.2%		4.7%		8%
3. Sensible initial steam, 2d cylinder	bot.	.1232 lb.	top	.1473 lb.	mean	.1353 lb.
4. Sensible steam expanded into 2d cylinder	,,	.2111	,,	.1790	,,	.1950
Difference of Nos. 3 and 4		.0879		.0317		.0597
Do. per cent of No. 4		41.6%		17.7%		30.6%
Difference of Nos. 1 and 4		.0982 lb.		.0579 lb.		.0780 lb.
Do. per cent of No. 4		46.5%		32.3%		40%

	1st Cylinder.	2d Cylinder.
5. Mean units of work-heat per stroke by expansion	13.85 units.	15.75 units.
6. Of which are provided by freed constituent heat	3.48 ,,	1.59 ,,
7. And by condensation of expanding steam	10.37 ,,	14.16 ,,
8. Weight of steam thus condensed	.0106 lb.	.0140 lb.
9. To be added to the mean sensible expanded steam	.1271 ,,	.1950 ,,
10. Total steam in evidence: sum of Nos. 8 and 9, the weights of condensed steam in both cylinders being added for the second cylinder	.1377 ,,	.2196 ,,
11. Deduct compressed steam	.0181 ,,	.0072 ,,
12. Net steam in evidence, consumed per stroke	.1196 ,,	.2124 ,,
13. Effective mean pressure, per square inch	16.60 ,,	4.10 ,,
14. Turns per minute	46.5 turns.	46.5 turns.
15. Speed of piston, in feet per minute	279 ft.	279 ft.
16. Indicator horse-power	30.03 H.P.	24.67 H.P.
17. Combined indicator horse-power	54.70 H.P.	
18. Net steam in evidence, consumed per hour, in cylinders	667.4 lbs.	1185 lbs.
19. Do. do. do. do. per H.P.	12.20 ,,	21.67 ,,
20. Effective mean pressure, per square inch, for the whole trial, lasting 10 hours. Speed as above stated	17.65 ,,	4.11 ,,
21. Average indicator power	32.03 H.P.	24.85 H.P.
22. Combined indicator horse-power	56.88 H.P.	
23. Water as steam consumed from the boiler per hour	1169.1 lbs.	
24. Do. do. do. do. per H.P.	20.55 ,,	
25. Of which there was condensed in the jacket, per H.P.	1.79 ,,	
26. And consumed within the cylinders, per H.P.	18.76 ,,	

Here is a contrast with the conditions and the performance of the factory engine as steam-jacketed previously discussed. The distribution of the expansion between the first and second cylinders is materially different, taking place mostly in the second cylinder in the factory engine, with its large cooling surface; and mostly in the first cylinder in the Hele engine. Whilst the steam, which is expanded in the first cylinder of the Hele engine down to 10 lbs. pressure, dropped only 3 lbs. and 2½ lbs. in pressure, in the intermediate space, in the factory engine the steam, which is expanded down to above 20 lbs. pressure in the first cylinder, drops 10 lbs. and 11 lbs. in the intermediate space. There is evidence of much greater condensation in the intermediate space, including the clearance of the second cylinder, in the factory engine, than in the Hele engine; since the drop of pressure is much greater in the former than in the latter, and in the former is much greater than that which would be due to simple enlargement of volume. As the range of expansion in the second cylinder of the factory engine is greater than in that of the Hele engine, the initial condensation in the former—42.75 per cent—is greater than in the latter—30.6 per cent—and gives rise naturally to the greater drop of pressure in the factory engine. In the Hele engine, it is true, there is nearly twice as much initial condensation and re-evaporation,—comparing the initial steam of the first cylinder with the final steam of the second cylinder,—as there is in the factory engine. This arises from the much greater ratio of combined working expansion in the former than in the latter—13.185 against 9.276. It follows naturally that the Hele engine performed its work more economically than the factory engine, as follows:—



Figs. 240.—B. Donkin & Co.'s Compound Tandem Steam Engine at Hele:—Sample Indicator Diagrams.

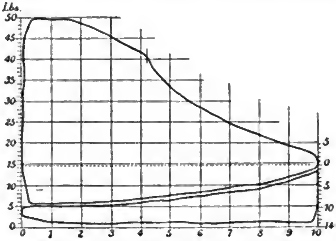
	Factory.	Hele.
Steam through the cylinders per horse-power, in evidence by sample indicator diagrams	20.30 lbs.	21.67 lbs.
Do. do. from the boiler, averaged for the whole period of trial:—In cylinders	18.50	18.76
Condensed in jackets	3.82	1.79
Total......................................	22.32	20.55

In both cases the steam in the cylinders brought into evidence by the sample indicator diagrams is greater than that actually measured for the whole period from the boilers into the cylinders. The difference, as was before suggested, may arise from exceptional conditions at the time when the diagrams were taken. But it is remarkable that, whilst the total steam consumed per horse-power is less in the Hele engine than in the factory engine, the difference in favour of the Hele engine arises from the lower rate of consumption of steam in the jacket.

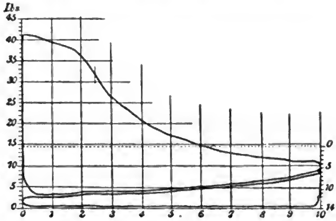
It may be noted that the steam-jacket was applied to both of the factory cylinders, and was applied only to the second cylinder of the Hele engine. The results of performance appear to indicate at first sight that the jacket on the first cylinder was not necessary for the purpose of economy. The indication is seeming only, since, for the work of expansion, the two cylinders may be taken as one, and the jacketing of the first, as of the second, Hele cylinder would no doubt have proved of benefit.

Upon the whole, it appears that the comparatively greater efficiency of the Hele engine is not due so much to the greater ratio of expansion adopted in it, as to the better proportions of the cylinders, in themselves, and as between the first and second cylinders.

Woolf compound rotative pumping engines were erected by Messrs. Simpson & Co., at the Lambeth Water-works, Thames Ditton.[1] They are on the Woolf principle, with a beam, and vertical cylinders at one end of the beam, and a pump and a crank and fly-wheel at the other end. There are four such engines, side by side, working in pairs on one crankshaft. The cylinders are steam-jacketed at the sides and the covers at the top, but not at the bottoms. They are worked with a piston-valve placed between and common to both. The first cylinder is 28 inches in diameter, with a stroke of 5 feet 6⅜ inches; the second is 46 inches by 8 feet; and the net capacities are 23.660 and 92.328 cubic feet, in the ratio of 1 to 3.904. The total clearance of the first cylinder is .644 cubic foot, or 2.72 per cent of the stroke; and that of the second cylinder is 2.127 cubic feet, or 2.30 per cent of the stroke. The intermediate space is 2.275 cubic feet, traversing the



From top of 1st Cylinder, and bottom of 2d Cylinder.



From bottom of 1st Cylinder, and top of 2d Cylinder.

Figs. 241.—Woolf Engines, Lambeth Water-works: Indicator Diagrams.

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[1] "On Double-cylinder Pumping Engines," by David Thomson, in the *Proceedings of the Institution of Mechanical Engineers*, 1862; page 259.

piston-valve, and the total space, including cylinder clearances—from piston to piston—at the beginning of the stroke, is (2.275+.644+2.127=) 5.046 cubic feet. The initial volume for expansion in the second cylinder is (23.66+5.046=) 28.706, and the volume of the expanded steam at the end of the stroke of the second cylinder is (92.328+5.046=) 97.374 cubic feet.

The effective pressure in the boiler was 40 lbs. per square inch. The indicator diagrams, figs. 241, were taken whilst the engine was steadily at work, making 14 turns per minute. The steam from the boiler is cut off in the first cylinder at 43 per cent at the top, and at 25 per cent at the bottom.

Total pressure, first cylinder:—Cut off:	top,	40 lbs.;	bottom,	32 lbs.
Do. do. Expanded:	,,	18	,,	11.2
Do. second cylinder:—Initial:	bottom,	13½	top,	8.7
Do. do. Expanded,	,,	5	,,	3
Total exhaust pressure for compression:—				
First cylinder	top,	6 lbs.;	bottom,	3½ lbs.
Second cylinder	bottom,	1 ,,	top,	1 ,,

The ratios of expansion in the first cylinder are as 10.817 to (23.66+.644=) 24.304 cubic feet for the top, or as 1 to 2.247; and as 6.558 to 24.304 cubic feet for the bottom, or as 1 to 3.706. The ratio of intermediate expansion, adjusted for the presence of residual steam in the interspace, is as 24.304 to 27.316 cubic feet, or as 1 to 1.124; and the ratio of expansion into the 2d cylinder is, for the top and the bottom, as 28.706 to 97.374 cubic feet, or as 1 to 3.392:—

Actual ratios of expansion:—						
(1) 1st cylinder	top,	2.247;	bot'm,	3.706;	mean,	2.976
(2) Intermediate	,,	1.124	,,	1.124	,,	1.124
(3) 2d cylinder	bot'm,	3.392	top,	3.392	,,	3.392
Total expansion-ratio		8.566		14.130		11.348
Net or working expansion-ratio		7.622		12.570		10.096
Ratios of initial and final pressures by expansion:—						
(1) 1st cylinder	top,	2.222;	bot'm,	2.857;	mean,	2.540
(2) Intermediate	,,	1.333	,,	1.287	,,	1.310
(3) 2d cylinder	bot'm,	2.700	top,	2.900	,,	2.800
Total ratio		7.997		10.660		9.329
Net or working ratio		6.000		8.285		7.143

The following are the quantities of sensible steam at different points of the system:—

1. Sensible steam cut off, 1st cylinder	top,	1.0535 lbs.;	bot'm,	.5175 lb.;	mean,	.7855 lb.
2. Do. expanded do.	,,	1.1155	,,	.7145	,,	.9150
Difference of Nos. 1 and 2		.0620		.1970		.1295
Do. per cent of No. 2		5.62%		27.57%		14.15%
3. Sensible steam, initial, 2d cylinder	bot'm,	1.0050 lbs.;	top,	.6631 lb.;	mean,	.8340 lb.
4. Do. expanded, do.	,,	1.3436	,,	.8276	,,	1.0856
Difference of Nos. 3 and 4		.3386		.1645		.2516
Do. per cent of No. 4		25.20%		19.88%		23.18%
Do. of Nos. 1 and 4		.2901 lb.		.3101 lb.		.3001 lb.
Do. per cent of No. 4		21.59%		37.47%		27.64%

	1st Cylinder.	2d Cylinder.
5. Mean units of work-heat per stroke by expansion.........	59.11 units	104.19 units
6. Of which are provided by freed constituent heat..........	13.68 ,,	18.91 ,,
7. And by condensation of expanding steam....................	45.43 ,,	85.28 ,,
8. Weight of steam thus condensed..............................	.0470 lb.	.0848 lb.
9. To be added to the mean sensible expanded steam.......	.9150 ,,	1.0856 lbs.
10. Total sum in evidence: sum of Nos. 8 and 9, in which the condensed steam of both cylinders is taken in for the 2d cylinder..	.9620 ,,	1.2174 ,,
11. Deduct compressed steam..	.0179 ,,	.0097 lb.
12. Net steam in evidence consumed per stroke...............	.9441 ,,	1.2077 lbs.
13. Effective mean pressure per square inch.................. ..	21.28 lbs.	5.04 ,,
14. Turns per minute..	14 turns	14 turns
15. Speed of piston, in feet per minute..........................	155 feet	224 feet
16. Indicator horse-power..	61.51 H.P.	56.85 H.P.
17. Combined indicator horse-power.............................	118.36 H.P.	
18. Net steam in evidence by the indicator, consumed per hour, in cylinders..	1586 lbs.	2030 lbs.
19. Do. do. do. per horse-power..........	13.40 lbs.	17.15 lbs.
20. Steam condensed in the jackets per hour....................	5 lbs.; or per H.P. .04 lb.	
21. Total steam in evidence by the indicator (2d cylinder) and the jackets, per hour: sum of Nos. 18 and 20...	2035 ,,	,, 17.19 lbs.
22. Water as steam delivered from the boiler, per hour......	2678 ,,	,, 22.62 ,,
23. Difference of Nos. 21 and 22..................................	643 ,,	,, 5.43 ,,
24. Do. per cent of No. 22..................................	24 per cent	

Here it appears that the steam jackets were ineffective, having condensed only 5 lbs. of steam per hour. The engine may therefore be considered as one having unjacketed cylinders; and accordingly it appears that there is as much as 24 per cent of the whole water as steam delivered from the boiler, permanently condensed in passing through the engine. The cylinders stood directly upon the condenser, separated only by the metal of the bottoms of the cylinders, 1½ inches or 2 inches in thickness, and probably much of the steam was condensed on their surfaces.

Receiver compound beam rotative engines, constructed by Messrs. Simpson & Co., were erected at the Lambeth Water-works, Thames Ditton.[1] They work in pairs at right angles on one crank-shaft. They are fitted with an intermediate receiver jacketed with steam from the boiler, on the system of Mr. E. A. Cowper, between the 1st and 2d cylinders. Both of the cylinders are steam-jacketed from the boiler, excepting the cover of the 1st cylinder; the receiver is steam-jacketed, except the cover. The condensation-water in the jackets returns by gravitation to the boilers. The steam is distributed by slide-valves, one valve at the top and one at the bottom, with adjustable expansion-plates on the back. A jet condenser is used. The 1st cylinder is 21 inches in diameter, the 2d cylinder is 36 inches, and the stroke for each cylinder is 5½ feet. The net capacities of the cylinders are 13.227 cubic feet and 38.877 cubic feet respectively, being in the ratio of 2.939. The total clearance of the 1st cylinder is .215 cubic foot, or 1.65 per cent of the net capacity, at each end. That of the 2d cylinder is 1.035 cubic feet, or 2.68 per cent. The net capacity of the intermediate receiver, including its connections, is equal to that of the 2d cylinder.

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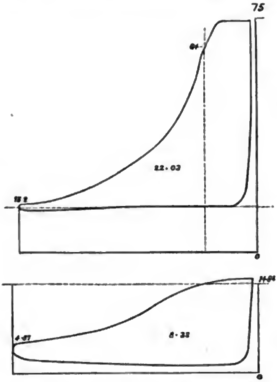
[1] See Mr. Mair's paper "On the Independent Testing of Steam Engines," already referred to. The engine here noticed is engine J in Mr. Mair's paper.



Figs. 242.—Pumping Steam Engines, by Simpson & Co., Lambeth Water-works, at Thames Ditton (J engine): Average Indicator Diagrams.

In the test made by Mr. Mair the average period of admission in the 1st cylinder, according to the average diagrams, figs. 242, is 21 per cent of the stroke, for which the initial volume for expansion in the 1st cylinder, including clearance, is 2.993 cubic feet, and the final volume is 13.442 cubic feet; for which the ratio of expansion is 4.491. The initial volume for expansion in the 2d cylinder, in which the steam is cut off at, say, 43.3 per cent, is 17.87 cubic feet; and the final volume is 39.912 cubic feet, making the working ratio of expansion 2.233. But the total ratio of expansion from the 1st into the 2d cylinder is the ratio of their capacities plus their respective clearances—13.442 and 39.912 cubic feet, or 2.947. The intermediate ratio of expansion is therefore (2.947 ÷ 2.233 =) 1.320.

The effective pressure in the boiler was 61 lbs., and the total pressure was 76 lbs. per square inch. The total initial pressure for expansion in the 1st cylinder is 64 lbs., and the final pressure is 15.2 lbs. In the 2d cylinder the initial pressure for expansion is 10 lbs., reduced to 4.67 lbs. per square inch at the end of the stroke. The total exhaust-pressure for compression in the 1st cylinder is 15 lbs., in the 2d cylinder, 1½ lbs. per square inch.

Actual ratios of expansion:—	
(1) 1st cylinder	4.491
(2) Intermediate	1.320
(3) 2d cylinder	2.233
Total expansion-ratio	13.236
Net do. do.	10.031
Ratios of initial and final pressures by expansion:—	
(1) 1st cylinder	4.217
(2) Intermediate	1.520
(3) 2d cylinder	2.141
Total ratio	13.724
Net or working ratio	9.029

The quantities of steam made sensible to the indicator are as follows:—

1. Sensible steam cut off, 1st cylinder	.4530 lb.
2. Do. do. expanded, do.	.5308
Difference of Nos. 1 and 2	.0778
Do. per cent of No. 2	14.66°/₀
3. Sensible steam cut off, 2d cylinder	.4718 lb.
4. Do. do. expanded, do.	.5188
Difference of Nos. 3 and 4	.0470
Do. per cent of No. 4	9.06°/₀
Do. of Nos. 1 and 4	.0658 lb.
Do. per cent of No. 4	12.68°/₀

	1st Cylinder.	2d Cylinder.
5. Work-heat per stroke by expansion	56.14 units	27.05 units
6. Of which are provided by freed constituent heat	13.48 „	5.39 „
7. And by condensation of expanding steam	42.66 „	21.66 „
8. Weight of steam thus condensed	.0443 lb.	.0216 lb.
9. To be added to the sensible expanded steam	.5308 „	.5188 „
10. Total steam in evidence: sum of steam condensed in both cylinders, line 8, being added to No. 9 for 2d cylinder	.5751 „	.5847 „
11. Deduct compressed steam	.0587 „	.0088 „
12. Net steam in evidence consumed per stroke	.5164 „	.5759 „
13. Effective mean pressure per square inch	22.01 lbs.	8.18 lbs.
14. Turns per minute	23.98 turns	23.98 turns
15. Speed of piston, in feet per minute	264 feet	264 feet
16. Indicator horse-power	60.93 H.P.	66.47 H.P.
17. Combined indicator horse-power	127.40 H.P.	
18. Net steam in evidence consumed per hour in cylinders	1486 lbs.	1657 lbs.
19. Do. do. do. per horse-power	11.66 lbs.	13.01 lbs.
20. Steam condensed in the jackets per hour, actually observed	256 lbs.; or per H.P. 2.02 lbs.	
21. Total steam in evidence in the 2d cylinder, by the indicator and the jackets, per hour	1913 „	„ 15.03 „
22. Dry steam actually delivered per hour from the boiler, and passed through the cylinders	1679 „	„ 13.17 „
23. Do. do. do. condensed in jackets	256 „	„ 2.02 „
24. Total steam actually delivered per hour from the boiler: sum of Nos. 22 and 23	1935 „	„ 15.19 „
25. Excess over total steam in evidence by the indicator, plus the steam condensed in the jackets (No. 21)	22 „ or 1 per cent	
26. Priming water delivered from the boiler, directly observed	70 „ or per H.P. .55 lbs.	
27. Total steam and primed water delivered from the boiler, directly observed	2005 „	„ 15.74 „
28. Coal consumed per indicator horse-power per hour	1.55 lbs. to 1.60 lbs.	

Herein is a noteworthy agreement, the virtual identity of the quantities of steam as brought into evidence passed through the cylinders by the indicator diagram, and as directly measured from the boiler. They are respectively 13.01 lbs. and 13.17 lbs., and, adding the weight of steam condensed in the jackets, the quantities are 15.03 lbs. and 15.19 lbs., differing only by 1 per cent. The engines were independently tested by Mr. E. A. Cowper in October, 1881, for a period of 24 hours, when the two pairs of engines exerted a combined power of 240.12 horse-power, or 120.06 horse-power for each pair. He states in his report,[1] that the quantity of water fed into the boilers was 13.397 pounds per indicator horse-power per hour. This water comprised the steam passed through the cylinders and the priming; and allowing .55 lb. as before for priming, the remainder, (13.397−.55=) 12.847 lbs., is the weight of water as dry steam passed through the cylinders per indicator horse-power per hour, against 13.01 lbs. as deduced above from indicator diagrams, and 13.17 lbs. as actually delivered from the boilers. The steam-jacketing of the receiver acted beneficially, no doubt, in so minimizing the consumption of steam. The steam supplied to the jackets was condensed in the following proportions:—

1st cylinder.........	.036315	lb. per stroke, or	.82	lb. per	H.P.	per hour.
Receiver.............	.011609	,, ,,	.26	,,	,,	,,
2d cylinder.........	.041326	,, ,,	.94	,,	,,	,,
Total.........	.089250	,, ,,	2.02	,,	,,	,,

A *pair of compound rotative beam engines*, designed by Mr. E. Leavitt, jun., was erected for the water-works at Lawrence, Massachusetts, U.S.A.[2] The first and second cylinders of each engine, illustrated in Section IV. following, are connected, one to each end of the beam. They are placed together under the main centre, and directed obliquely, each to its proper end of the beam, the lower ends of the cylinders being close together, and the upper ends lying apart. The steam from the first cylinder is exhausted direct into the corresponding end of the second cylinder; and thence to the condenser. One end of the beam is connected to the fly-wheel shaft, and the other end to the pump. The cylinders are thoroughly steam-jacketed; they are fitted with gridiron valves, having large area of opening, and short movement. The steam-valves of the first cylinder are worked by means of cams. The cylinders are 18 inches and 38 inches in diameter, with 8 feet of stroke; having net capacities of 14.136 cubic feet and 63.016 cubic feet respectively, in the ratio of 1 to 4.458. The clearance-space for the first cylinder is 2.56 per cent of the net capacity at the top, and 2.31 per cent at the bottom, or .362 cubic foot and .326 cubic foot respectively. For the second cylinder, they are 1.54 per cent and 1.82 per cent, or .970 cubic foot, and 1.147 cubic feet, for the top and the bottom respectively. The volume of the pipe connecting the upper ends of the cylinders is 1.402 cubic feet. The steam is cut off in the first cylinder at 31.4 per cent at the top, and 31.5 per cent at the bottom, making the initial volumes for expansion (4.439+.362 clearance=) 4.801 cubic feet and (4.453+.326 clearance=) 4.779 cubic feet; and the actual ratios of expansion in the first cylinder are, 3.020 and 3.030.

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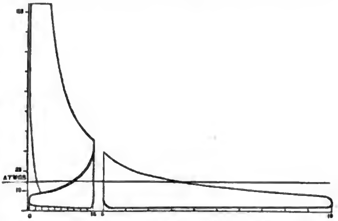
[1] *Engineering*, December 9, 1881; page 575.

[2] See *Reports on the Trial of Duty and Capacity of the Pumping Engines, Lawrence Water-works*, by Wm. E. Worthen, J. C. Hoadley, and J. P. Davis; May, 1876.



Figs. 243.—Compound Beam Rotative Pumping Engines, Lawrence, Massachusetts, designed by E. Leavitt:—Sample Indicator Diagrams.

The initial volumes for expansion into the second cylinder are (14.136+.362=) 14.498 cubic feet, and (14.136+.326=) 14.462 cubic feet, for the top and the bottom respectively; and the total final volumes for expansion, reckoned to the end of the stroke, are (63.016+.970 clearance+1.402 pipe=) 65.388 cubic feet, for the top; and (63.016+1.147=) 64.163 cubic feet for the bottom; for which the actual expansion-ratios are 4.510 and 4.437. But these ratios include intermediate expansion, whilst the intermediate space for expansion is to be reduced for the presence of residual steam. Adjusted accordingly, the expansion-ratios are as follows:—

Actual ratio of expansion:—	top	bottom	mean
(1) 1st cylinder,	3.020;	3.030;	3.025
(2) Intermediate,	1.124;	1.062;	1.093
(3) 2d cylinder,	4.013;	4.178;	4.096
Total expansion-ratio,	13.625	13.444	13.535
Net do.	12.122	12.660	12.391

For a run lasting 16 hours, with steam of 90 lbs. sensible pressure in the boiler, the engine worked at an average speed of 16.25 turns, or 260 feet of piston, per minute. Sample indicator diagrams are shown in figs. 243.

The absolute pressures of steam are as follows:—

	top	bottom
1st cylinder, cut off	100.2 lbs.	102.4 lbs.
„ expanded	33.8 „	34.9 „
„ initial pressure, return-stroke,	29.6 „	28.6 „
„ final pressure, back-stroke for compression	9.2 „	9.9 „
2d cylinder, initial	26.14 „	28.4 „
„ expanded	6.14 „	5.42 „

Ratio of initial and final pressures by expansion:—

	top	bottom
(1) 1st cylinder	2.990	2.900
(2) Intermediate	1.293	1.229
(3) 2d cylinder	4.257	5.240
Total ratio	16.458	18.675
Net or working ratio	12.728	15.200

The weights of sensible steam are as follows:—

	top	bot.	mean
1. 1st cylinder, cut off	1.1105 lbs.	1.1280 lbs.	1.1193 lbs.
2. „ expanded	1.2150 „	1.2380 „	1.2265 „
Difference of Nos. 1 and 2	.1045	.1100	.1072
„ per cent of No. 2	8.60%	8.88%	8.74%

3. 2d cylinder, initial	top	.9467	bot'm	1.0195	mean	.9831 lb.
4. ,, expanded	,,	1.0920	,,	.9496	,,	1.0208 lbs.
Difference of Nos. 3 and 4		.1453		–.0699		.0377
,, per cent of No. 4		13.30°/。		–7.36°/。		3.69°/。
,, of Nos. 1 and 4		–.0185 lb.		–.1784 lb.		–.0984 lb.
,, per cent of No. 4		1.69°/。		–18.79°/。		–9.73°/。

	1st Cylinder.		2d Cylinder.	
5. Mean work-heat per stroke by expansion	108.60 units.		164.50 units.	
6. Of which is provided by freed constituent heat	26.49 ,,		23.69 ,,	
7. And by condensation of expanding steam	82.11 ,,		140.81 ,,	
8. Weight of steam thus condensed	.0884 lb.		.1414 lb.	
9. To be added to the sensible expanded steam	1.2265 ,,		1.0208 ,,	
10. Total steam in evidence: sum of steams condensed in both cylinders being added to No. 9 for 2d cylinder	1.3149 ,,		1.2506 ,,	
11. Deduct compressed steam	.0731 ,,		.0104 ,,	
12. Net steam in evidence consumed per stroke	1.2418 ,,		1.2402 ,,	
13. Effective mean pressure per square inch	53.33 lbs.		10.20 lbs.	
14. Turns per minute	16.25 turns.		16.25 turns.	
15. Speed of piston in feet per minute	260 feet.		260 feet.	
16. Indicator horse-power	106.90 H.P.		91.14 H.P.	
17. Combined indicator horse-power	198.04 H.P.			
18. Net steam in evidence consumed per hr. in cylinders	2422 lbs.		2418 lbs.	
19. Do. do. per horse-power	12.20 ,,		12.19 ,,	
20. Steam condensed in the jackets per hour	338.85 ,,	or	1.71 ,,	per H.P.
21. Total steam in evidence by the indicator, in the 1st cylinders, and the jackets, per hour	2760.85 ,,	or	13.91 ,,	,,
22. Total water as steam delivered from the boiler[1]	2796 ,,	or	14.06 ,,	,,
23. Coal, Cumberland, consumed per H.P. per hour			1.69 ,,	,,

Here there is a very nearly exact coincidence between the quantity of water as steam, No. 21, brought into evidence by the indicator, plus the condensation water from the jacket, and the total water as steam delivered from the boiler—about 14 lbs. per indicator horse-power per hour.

The calculation of heat converted into work during expansion, No. 8, was only approximative, being based on indicator diagrams taken on the last day of the trials. It is seen, line 12, that the net weights of steam in evidence, consumed per stroke, are nearly identical for the two cylinders.

It is seen that the ratios of the initial and final pressures in the 1st cylinder are nearly identical with the actual ratios of expansion. In the 2d cylinder at the top, also, they are nearly alike, but at the bottom the ratio of pressures is materially greater than that of expansion. In three of the four situations the expansion is thus seen to follow sensibly the hyperbolic law. The final pressures in the 2d cylinder, as observed, were 6.14 lbs. and 5.42 lbs. According to the hyperbolic law, reckoned on the gross ratios of expansion, they would be 7.75 lbs. and 7.70 lbs.

A *horizontal compound receiver-engine*, constructed by J. & E. Wood, Bolton, and erected at the mills of F. B. Booth & Co., Leigh, near Manchester. The cylinders, figs. 244, are 27 inches and 42 inches in diameter, with a stroke of 5 feet, working to cranks at right angles on the fly-wheel shaft. The intermediate receiver is of plate-iron, cylindrical, 3 feet in diameter, and 10 feet long. The cylinders, receiver, and pipes are lagged, but not jacketed. The first cylinder is fitted with Wood's Corliss gear, and the second with a slide-valve.[1] The net capacity of the 1st cylinder is 19.88 cubic feet, and that of the 2d is 48.10 cubic feet. The ratio of the capacities is 2.420. The total clearance of the 1st cylinder is 1.93 per cent, or .383 cubic foot at each end; and that of the 2d cylinder is 8.34 per cent, or 4.010 cubic feet at each end. The intermediate receiver and passages together have a capacity of 93.32 cubic feet. This is unusually large, and the object of its being constructed so large was to admit of the continuous collection of condensation-water for a considerable period of time. For one observation, illustrated by the indicator diagrams, figs. 245, the steam is cut off at 24 per cent at the front, and 23 per cent at the back; and at 50 per cent in the 2d cylinder, for which the gross volumes, including clearances, are respectively 5.155 cubic

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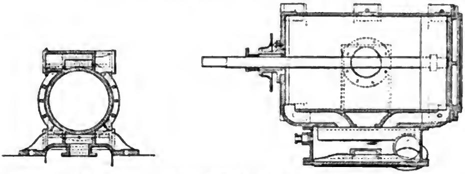
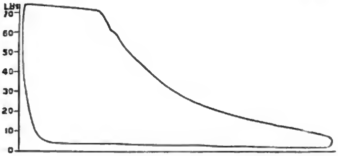
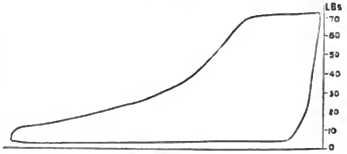
[1] The destination of the condensation-water in the steam jackets is not specifically stated in the report; but it is inferred from the language of the report, page 37, that that water was not returned to the boiler.



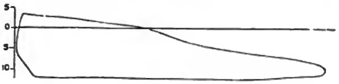
Figs. 244.—J. & E. Wood's Compound Receiver Steam Engine: Cylinders. Scale 1/56th.



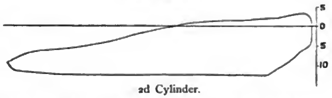
1st Cylinder.



1st Cylinder.



2d Cylinder.



2d Cylinder.

Figs. 245.—Wood's Horizontal Receiver Compound Engine:—Sample Indicator Diagrams.

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[1] See *Engineering*, November 21, 1873, vol. xvi. page 421, for an account, with illustrations, of this engine.

feet and 4.956 cubic feet for the 1st cylinder, or a mean of 5 cubic feet; and 28.062 cubic feet for the 2d cylinder. The final volumes by expansion, reckoned to the ends of the strokes, are 20.263 cubic feet and 52.110 cubic feet. The intermediate expansion took place from the volume 20.263 cubic feet to 28.062. The expansion-ratios, therefore, are as follows:—

Actual ratio of expansion:—	(1) 1st cylinder	4.053
	(2) Intermediate	1.385
	(3) 2d cylinder	1.857
	Total expansion-ratio	10.423
	Net or working ratio, (1) and (3)	7.526

The pressure of steam in the boiler was 77 lbs. per square inch above the atmosphere, or 92 lbs. absolute. The absolute pressures of steam in the cylinders are as follows:—

1st cylinder, cut off	front, 87 lbs.;	back, 84 lbs.
,, expanded	,, 23 ,,	,, 24.2 lbs.
2d cylinder, cut off	front, 12 lbs.;	back, 14 lbs.
,, expanded	,, 7 ,,	,, 8 ,,

Ratio of initial and final pressures by expansion:—

(1) 1st cylinder	front, 3.783	back, 3.471	mean, 3.627
(2) Intermediate	,, 1.917	,, 1.730	,, 1.824
(3) 2d cylinder	,, 1.714	,, 1.750	,, 1.732
Total ratio	12.430	10.508	11.458
Net or working ratio, (1) and (3)	6.484	6.074	6.279

The exhaust-pressure for compression is 20 lbs. per square inch for the 1st cylinder, and 2.7 lbs. and 3.20 lbs. for the 2d cylinder.

The weights of sensible steam, according to the indicator diagram and other particulars of performance, are as follows:—

1. 1st cylinder, cut off	front 1.0435 lbs.	back .9699 lbs.	mean 1.0067 lbs.
2. ,, expanded	,, 1.1753 ,,	,, 1.2280 ,,	,, 1.2016 ,,
Difference of Nos. 1 and 2	.1318 ,,	.2581 ,,	.1950 ,,
,, per cent of No. 2	11.08%	21.02%	16.23%
3. 2d cylinder, cut off	front .8811 lb.	back 1.0158 lbs.	mean .9484 lb.
4. ,, expanded	,, .9849 ,,	,, 1.1152 ,,	,, 1.0501 ,,
Difference of Nos. 3 and 4	.1038 ,,	.0994 ,,	.1016 ,,
,, per cent of No. 4	10.64%	8.41%	9.67%
,, of Nos. 1 and 4	−.0586 lb.	.1453 lb.	.0434 lb.
,, per cent of No. 4	−5.94%	12.58%	4.13%

	1st Cylinder.	2d Cylinder.
5. Mean work-heat per stroke by expansion	134.16 units.	111.00 units.
6. Of which is provided by freed constituent heat	29.32 ,,	8.30 ,,
7. And by condensation of expanded steam	101.84 ,,	102.70 ,,
8. Weight of steam thus condensed	.1075 lb.	.1034 lb.
9. To be added to the sensible expanded steam	1.2016 ,,	1.0501 ,,
10. Total steam in evidence: sum of steams condensed in both cylinders being added to No. 9 for 2d cylinder	1.3091 ,,	1.2610 ,,
11. Deduct compressed steam	.1203 ,,	.0753 ,,

	1st Cylinder.	2d Cylinder.
12. Net steam in evidence consumed per stroke	1.1888 lbs.	1.1857 lbs.
13. Effective mean pressure per square inch	32.45 ,,	9.80 ,,
14. Turns per minute	42 turns.	42 turns.
15. Speed of piston, in feet per minute	420 feet.	420 feet.
16. Indicator horse-power	236 H.P.	172.5 H.P.
17. Combined indicator horse-power	408.5 H.P.	
18. Net steam in evidence consumed per hour	5992 lbs.	5976 lbs.
19. Do. do. per horse-power	14.67 lbs.	14.63 lbs.

When the indicator diagrams were taken the engine had been running for twenty-six weeks, and was reported to give an average of 405 indicator horse-power, with an average consumption of 29 tons of coal per week of 60 hours, or at the rate of 2.67 lbs. per horse-power per hour. After making a deduction of one-fifth for heating purposes, the remainder, 2.14 lbs., is estimated as the net consumption of coal on the work of the engine per horse-power per hour. Messrs. Wood state that they ascertained that 8 lbs. of water was evaporated per pound of coal consumed. Thence, the quantity of water as steam consumed by the engine would be $(8 \times 2.14 =)$ 17.12 lbs. per horse-power per hour, which is 2.45 lbs. in excess of the quantity of steam (line 19) brought into evidence by the indicator diagram. These estimated results are much better than others which have been verified under like conditions, from other engines, and they need to be corroborated. The case is of some interest as one of non-jacketed compound cylinders.

Water was collected in the receiver at the rate of 576 lbs. per hour, or at the rate of $(576 \div 408.5 =)$ 1.41 lbs. per indicator horse-power. This water was, of course, discharged from the first cylinder, and is at the rate of .1143 lb. per stroke, which happens to be very little more than that of the weight of steam condensed for work during expansion.

The capacity of the receiver and passages was said to be 93.32 cubic feet. This is equal to nearly double the capacity of the second cylinder, and obviously is large enough, since the exhaust-lines of the indicator diagrams from the first cylinder are straight. When the receiver, of which the net capacity is 70.7 cubic feet, was half filled with water, and so reduced the volume for steam to $(93.32 - 35.35 =)$ 57.97 cubic feet, the form of the diagrams remained unaffected by the reduction in volume. The volume as reduced was 1.205 times the capacity of the second cylinder.

A *Woolf condensing beam-engine*, at the Florida Spinning Mill, Ghent, manufactured by A. & E. Gilain,[1] was tested by M. Dwelshauvers and M. de Wilde, in September, 1877. The cylinders are not jacketed, but the steam is superheated on its passage from the first to the second cylinder. The first cylinder is 27 inches, and the second cylinder is 47.2 inches in diameter; with strokes respectively of 7 feet 5 inches and 9 feet 10 inches. The capacities are as 1 to 4.05. The engine made 30 turns per minute, or 445 feet of the first piston per minute. The steam was cut off at ¼th of the stroke of the first piston. Assuming a clearance of 5 per cent, the actual ratio of expansion in the first cylinder would be 3.8. The net capacities of the first and second cylinders are as 1 to 4.05, which is the measure approximately of the total expansion into the second cylinder; but, allowing an intermediate expansion-ratio of say, 1.20, the working ratio would be (4.05 ÷ 1.20 =) 3.38, and (3.8 × 3.38 =) 11.42 would on these conditions be the total net ratio of expansion. The indicator horse-power for the first cylinder is 284.9; and, for the second, 106.4; together, 391.3 horse-power. The consumption of water from the boiler is at the rate of 14.83 lbs. per horse-power.

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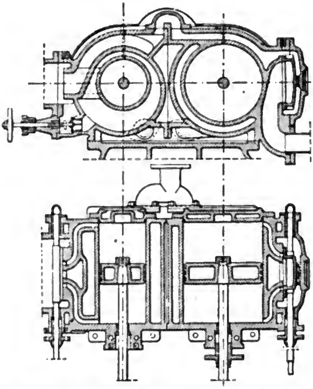
[1] *Bulletin de l'Association des Ingénieurs Sortis de l'École de Liége*, 1877; page 214.

A *compound condensing semi-portable engine*, constructed by Weyher and Richemond, at Pantin, and working at Herzog's Spinning Mills, Colmar, was tested in July, 1879, by MM. Walther-Meunier and G. Keller.[1] The two cylinders, figs. 246, are horizontal, fixed side by side, on the top of the boiler. A slide-valve is applied to each cylinder, and is placed laterally at the outer side. The first cylinder is steam-jacketed, except at the ends. The steam is distributed by means of an ordinary slide-valve, to each cylinder. The steam exhausted from the first cylinder is passed over the outside of the first jacket, being heated by the steam therein, and over the second cylinder to its valve-chest. The second cylinder is jacketed by the heated exhaust-steam on its upper part, and on half of its circumference by steam direct from the boiler. The ends are not jacketed. The cylinders are connected to two cranks at right angles on the main shaft. The steam from the boiler is admitted into the lower part of each steam-jacket, and from the jacket of the first cylinder it is conducted to its valve-chest. The experimentalists conclude that the whole of the condensation-water of the jackets was carried by the steam into and through the cylinders. The cylinders are 11.18 inches and 18.9 inches in diameter, with a stroke of 18.9 inches. The ratio of the capacities is 2.84. The clearances are respectively 6.4 per cent and 5 per cent of the net capacities of the cylinders. The average results of two days' trials show that the net pressure in the boiler was 95.5 lbs. per square inch, and that the steam was cut off at 45 per cent and 35 per cent of the stroke at the two ends respectively of the first cylinder, with 87 lbs. net, or 102 lbs. total initial pressure for expansion. The mean



Figs. 246.—Weyher and Richemond's Semi-portable Steam Engine:—Compound Cylinders. Scale 1/30th.

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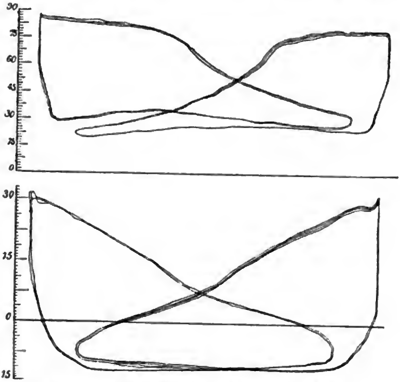
[1] *Bulletin de la Société Industrielle de Mulhouse*, 1880, page 5.

period of admission was 40 per cent, for which the actual ratio of expansion was 2.27, whilst that into the 2d cylinder was approximately 2.84, making a total working ratio of 6.46. Sample indicator diagrams are given, figs. 247. The capacity of the space answering to the receiver is very limited, and the exhaust-line of the indicator diagrams shows a rise of pressure towards the middle of the stroke, succeeded by a fall of pressure, as steam passes off to the second cylinder. There is no drop of pressure between the cylinders, the maximum admission pressure in the second cylinder being sensibly equal to the final pressure in the first cylinder; but the steam is very much wire-drawn in the second cylinder. The average speed was 88.5 turns per minute, making 78 indicator horse-power; the consumption of water from the boiler was at the rate of 16.81 pounds per indicator horse-power per hour, and that of dry steam was at the rate of 16.37 pounds.



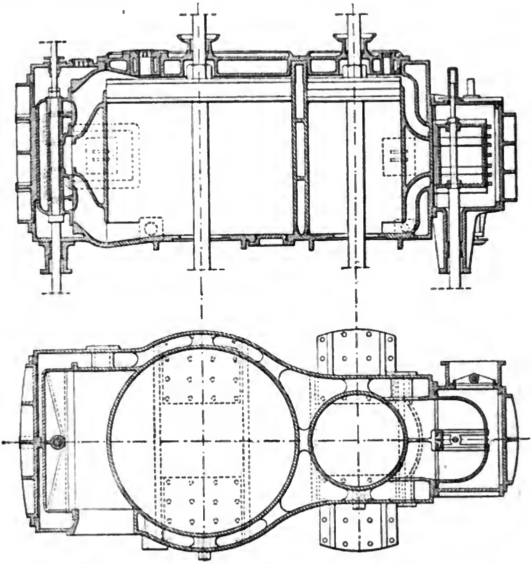
Figs. 247.—Weyher and Richemond's Semi-portable Engine:—Indicator Diagrams.

In another trial of this engine, cutting off in the first cylinder at 25½ per cent, expanding there 3⅓ times, and in the two cylinders together ($3^1/_3 \times 2.84 =$) 9.47 times; the engine made 90 turns per minute, and 63.86 horse-power, consuming from the boiler 16.94 pounds per indicator horse-power per hour; or, of dry steam, 16.03 pounds.

		1st Trial.	2d Trial.
Mean cut-off in the 1st cylinder		40°/₀	25°/₀
Do. do. do. 2d do.		41°/₀	41°/₀
Actual working ratio of expansion	1st cylinder	2.27	3.33
	2d do. (approximate)	2.260	2.260
	Both cylinders	5.130	7.526
Absolute maximum pressure in 1st cylinder, per square inch		87 lbs.	89 lbs.
Absolute pressure at cut-off, for expansion, ,, ,,		76 ,,	84 ,,
Absolute maximum pressure in 2d cylinder, ,, ,,		42.7 ,,	41 ,,
Absolute pressure at cut-off		28.5 ,,	28 ,,
Turns per minute		88.5 turns.	90 turns.
Indicator horse-power		78 H.P.	63.86 H.P.
Water as steam consumed per horse-power per hour		16.81 pounds.	16.94 pounds.
Dry steam do. do. do.		16.37 ,,	16.03 ,,

There is not a marked difference between the consumptions of water per horse-power for 40 per cent and 25 per cent admission into the first cylinder.

The *compound receiver-engines of the Liffey steam-vessel*, constructed by Messrs. Day, Summers & Co., have cylinders 36 inches and 72 inches in diameter, with 4 feet 2 inches of stroke. The ratio of the cylinders is 4, and the clearances are 8.22 per cent and 9.13 per cent of the capacity. The steam, 45 lbs. in the boiler, is cut off at 55 per cent in the first cylinder, and 45 per cent in the second cylinder. The actual ratios of expansion are:—

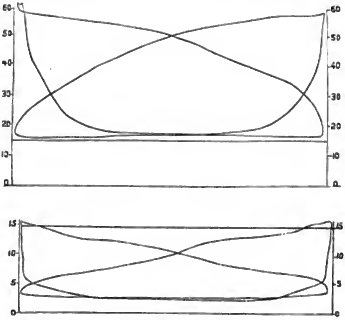


Figs. 248.—S.S. Liffey: Compound Cylinders, by Day, Summers & Co. Scale 1/48th.

In the 1st cylinder	1.715
,, 2d cylinder	2.220
Net combined ratio	3.807
Intermediate expansion	2.000
Gross expansion	7.614

The cylinders, figs. 248, are enveloped in the receiver, and are steam-jacketed at the upper side. The pressure of steam, according to the indicator diagrams, figs. 249, at the cut-off in the first cylinder, was 49½ lbs.

and 50½ lbs., expanded finally to 7 lbs. in the second cylinder, or to about one-seventh of the initial pressure—the gross expansion-ratio being 7.614. The indicator power is 409.5 in the first cylinder, 427.8 in the second, together 837.3 horse-power. The indicated quantity of steam per horse-power per hour was 14.74 pounds cut off in the first cylinder, and 17 pounds expanded in it; 10.75 pounds cut off in the second cylinder, and 11.99 pounds expanded in it. The coal, Cardiff, was reported to be consumed at the rate of 2.10 pounds per horse-power per hour.



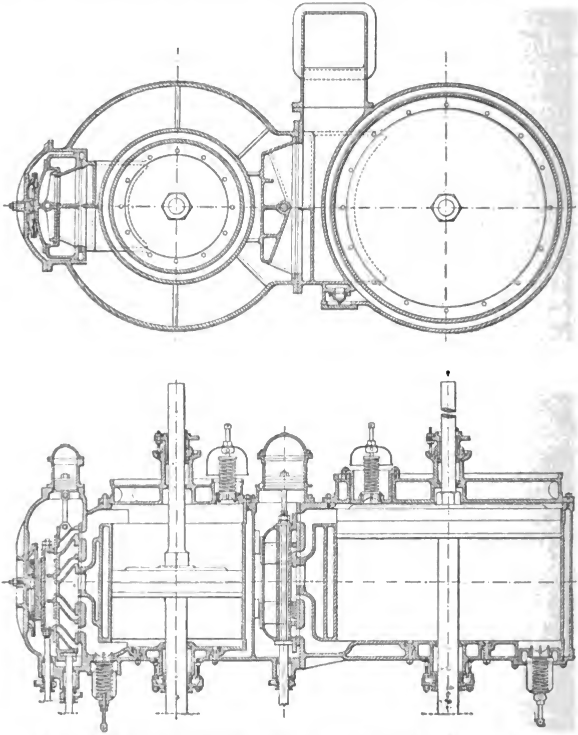
Figs. 249.—Engines of the Liffey, by Day, Summers & Co.:—Sample Indicator Diagrams.

The *compound receiver-engines of the Koning der Nederlanden*, constructed by Messrs. John Elder & Co., have two cylinders, 50 inches and 86 inches in diameter, with a stroke of 3½ feet, having capacities as 1 to 2.959. The cylinders, figs. 250, are steam-jacketed separately, and the receiver surrounds the first cylinder. The steam, according to the first set of diagrams, figs. 251, is cut off in the first cylinder at 52 per cent at the top, and 48 per cent at the bottom, with clearance-spaces of 9.9 per cent and 8.28 per cent respectively; in the second cylinder, at 55 per cent and 47½ per cent, with 5.65 per cent and 7.05 per cent clearances. The actual ratios of expansion are as follows:—

In the 1st cylinder	1.850
„ 2d cylinder	1.852
Net combined ratio	3.426
Intermediate expansion	1.360
Gross expansion	4.660

With 59 lbs. steam in the boiler, the initial pressure at cut-off in the first cylinder is 55 lbs. at the top, and 57 lbs. at the bottom. Steam of 24 lbs. and 25 lbs. is compressed at 7 per cent and 8 per cent of the stroke respectively. In the second cylinder the steam is cut off at 14½ lbs. and 16 lbs. pressure, and the exhaust is compressed at 10 per cent, at 3¼ lbs. and 2½ lbs. The indicator horse-power, at 65 turns per minute, is 775.60 and 996.18 in the cylinders respectively, together 1771.78 horse-power. The indicated quantity of steam per horse-power per hour is 14.45 lbs. cut off, and 14.85 lbs. expanded, in the first cylinder; and 13.14 lbs. cut off, and 13.28 lbs. expanded, in the second cylinder.

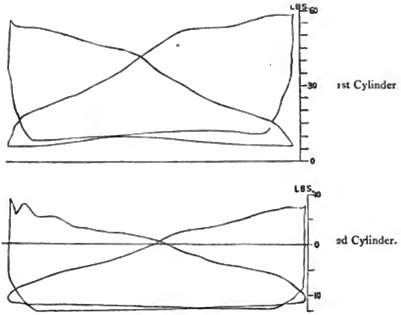
By the second set of indicator diagrams, figs. 251, taken at a speed of 61 turns per minute, the steam was cut off at 34½ per cent and 40 per cent at the top and the bottom of the first cylinder respectively; and in the second cylinder at 55 per cent and 47½ per cent as in the previous diagrams. The pressures at cut-off were 50 lbs. and 61½ lbs. in the first cylinder, and 10 lbs. and 13 lbs. in the second cylinder, expanded to 6½ lbs. The indicator horse-power was 674.33 in the first cylinder, and 720.78 in the second; together 1395.71 horse-power. The indicated steam cut off



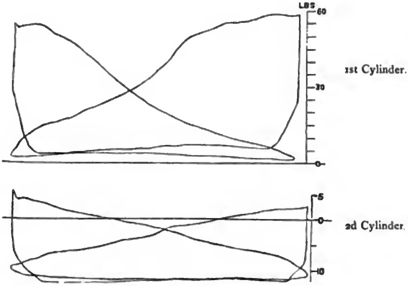
Figs. 250.—S.S. Koning der Nederlanden:—Compound Cylinders, by John Elder & Co. Scale 1/48th.

in the first cylinder was 11.85 lbs. per horse-power; expanded, 14.05 lbs.: cut off in the second cylinder, 12.34 lbs.; expanded, 13.40 lbs. The coal consumed was 1.61 lbs. per horse-power per hour. The actual ratios of expansion were as follows:—

In the 1st cylinder	2.358
„ 2d cylinder	1.852
Combined ratio	4.367
Intermediate expansion	1.314
Gross expansion	6.093



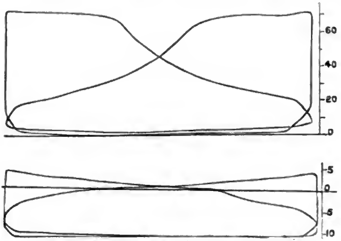
First Set.



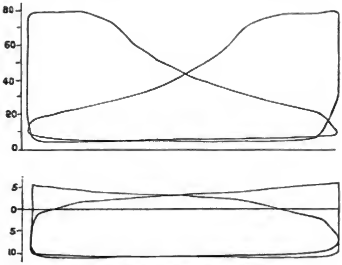
Second Set.

Figs. 251.—Engines of the Koning der Nederlanden:—Indicator Diagrams.

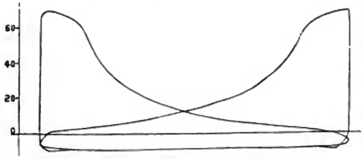
The condensing engine of the U.S. steamer Bache, having compound cylinders, was tested by Mr. C. E. Emery. He, in conjunction with Chief-engineer Loring, also tested the *compound engines of the U.S. steamer Rush; and the single-cylinder engines of the Dexter and the Dallas.*[1]



No. 3 Diagrams.—Bache: without Steam in Jacket.



No. 6 Diagrams.—Bache: Steam in Jacket.



No. 13 Diagrams.—Bache: 2d Cylinder only, without Steam in Jacket.

Figs. 252.—Indicator Diagrams from the Bache.

The Woolf engine of the Bache, designed by Mr. Emery, is of the steeple compound-cylinder type, the first cylinder, 16 inches in diameter, being placed over the second cylinder, 25 inches in diameter, and the pistons being fixed to one piston-rod, with a stroke of 2 feet. The steam from the upper cylinder passes to the valve-chest of the lower cylinder through a large pipe, regardless of the volume of the intermediate space—acting, in fact, as a receiver—as the distribution of the power between the two cylinders is regulated by means of an adjustable cut-off on the second cylinder. Steam is distributed to each cylinder by a short slide-valve at each end, and for both cylinders there are independently adjustable cut-off plates on the backs of the main valves. From this description, it appears that the engine, though nominally of the Woolf type, works as a receiver-engine. The net capacity of the second cylinder is 2.44 times that of the first cylinder. The second cylinder only is steam-jacketed—at the sides. The condensation-water from the jacket is returned to the boiler. The effective pressure in the boiler was 80 lbs. per square inch. Four series of experiments were conducted: 1st, without steam in the jacket, but draining the intermediate chamber; 2d, with steam in the jacket, and draining the intermediate chamber; 3d, with the second cylinder only in action, without steam in the jacket; 4th, the same, with steam-jacketing. Sample indicator diagrams are shown in figs. 252 and 253. The leading results are recorded in table No. 117, in which the actual ratios of expansion are stated, as given by Mr. Emery. The fuel, anthracite, was consumed at the rate of 8.84 pounds per square foot of grate; and 9.13 pounds of water per pound of coal was evaporated from 92.5° Fahr.

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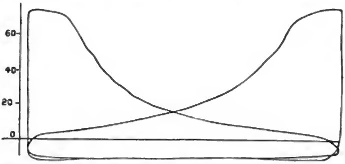
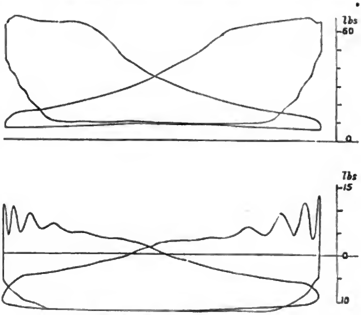
[1] *Journal of the Franklin Institute*, February and March, 1875; pages 105, 197. See also notices of the trials in the *Proceedings of the Institution of Civil Engineers*; vol. xl. page 292; and vol. xli. page 296.

The compound receiver-engines of the Rush have vertical cylinders, 24 inches and 38 inches in diameter, with a stroke of 27 inches; both of which are thoroughly steam-jacketed, felted, and lagged. The net capacities of the cylinders are as 1 to 2.509. The total clearance in the first cylinder is 7.89 per cent; and in the second, 5.85 per cent. Results of performance are given in table No. 117.

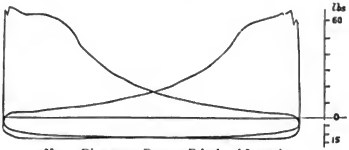
The single cylinder of the engine of the Dexter is inverted, 26 inches in diameter, with a 3-feet stroke, not jacketed, but thoroughly felted and lagged. The total clearance is 5.37 per cent. The single cylinder of the Dallas, also inverted, is 36 inches in diameter, with a stroke of 2½ feet; it is covered with non-conducting composition, and



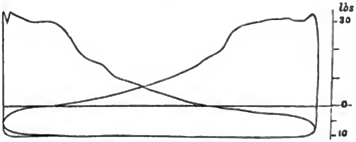
No. 16 Diagrams.—Bache: 2d Cylinder only, with Steam in Jacket.



No. 18 Diagrams.—Rush: Steam-jacketed.



No. 21 Diagrams.—Dexter: Felted and Lagged.



No. 29 Diagrams.—Dallas: Felted and Lagged.

Figs. 253.—Indicator Diagrams from the Bache, Rush, Dexter, and Dallas.

lagged. The total clearance is 8.02 per cent. The steam is distributed by a short slide-valve, with adjustable cut-off plates on the back of it. Results of performance of these two engines are given in table No. 117.

Table No. 117.

WORK OF EXPANDED STEAM—AMERICAN MARINE CONDENSING ENGINES OF THE BACHE, RUSH, DEXTER, AND DALLAS. May and August, 1874.

STEAMER: Condition of Cylinders. 1	Cut-off. 2	Actual Ratio of Expansion. 3	Total Maximum Pressure per square inch in 1st Cyl. 4	Turns per Minute. 5	Speed of Piston in feet per Minute. 6	Indicator Horse-power. 7	Weight of Steam per I.H.P. per Hour.		
							By Indicator 1st Cylinder. 8	By Indicator 2d Cylinder. 9	Actually Consumed. 10
	p. cent.	ratio.	lbs.	turns.	feet.	H.P.	pounds.	pounds.	pounds.
Bache (Woolf). (cylinders 16 in. and 25 in.; stroke 2 ft.)									
Without steam in jacket:—									
1. 1st cylinder	23.1	3.75	90	42.6	170.4	35.59	15.41	12.70	23.76
Both cylinders	—	9.15	—	—	—	55.93	—	—	—
2. 1st cylinder	33.5	2.73	91	47.7	190.8	49.85	15.67	12.31	23.04
Both cylinders	—	6.66	—	—	—	77.00	—	—	—
3. 1st cylinder	40.5	2.31	88	49.3	197.2	53.32	15.37	12.35	23.21
Both cylinders	—	5.63	—	—	—	85.81	—	—	—
With steam in jacket:—									
4. 1st cylinder	10.3	6.91	90.7	38.9	155.6	22.85	14.55	18.53	25.11
Both cylinders	—	16.85	—	—	—	46.40	—	—	—
5. 1st cylinder	22.9	3.77	92.1	48.2	192.8	40.87	14.10	15.76	20.71
Both cylinders	—	9.19	—	—	—	77.45	—	—	—
6. 1st cylinder	31.8	2.86	93.6	53.2	212.8	54.06	14.97	14.77	20.33
Both cylinders	—	6.98	—	—	—	99.18	—	—	—
7. 1st cylinder	39.6	2.35	90.4	56.3	225.2	58.09	15.85	15.25	20.37
Both cylinders	—	5.73	—	—	—	110.50	—	—	—
8. 1st cylinder	39.8	2.34	90.3	55.6	222.4	60.56	15.85	16.00	21.97
Both cylinders	—	5.71	—	—	—	104.03	—	—	—
9. 1st cylinder	45.2	2.09	89.5	53.6	214.4	55.44	14.85	14.53	22.38
Both cylinders	—	5.10	—	—	—	102.26	—	—	—
10. 1st cylinder	55.4	1.74	87.4	60.0	240.0	65.91	17.27	16.21	21.17
Both cylinders	—	4.24	—	—	—	134.53	—	—	—
Without steam in jacket:—									
11. 2d cylinder only	4.75	11.82	89.3	37.3	149.2	47.24	21.03	—	35.08
12. Do. do.	9.6	7.62	91.1	44.9	179.6	71.75	17.76	—	29.62
13. Do. do.	15.5	5.32	87.7	47.1	188.4	89.14	17.35	—	26.25
With steam in jacket:—									
14. 2d cylinder only	4.2	12.62	89.2	39.9	159.6	54.84	16.42	—	27.11
15. Do. do.	8.1	8.57	90.5	46.2	184.8	74.62	15.58	—	24.09
16. Do. do.	16.3	5.11	91.1	53.8	215.2	116.01	16.25	—	23.15
17. Do. do.	43.7	2.18	42.0	45.3	181.2	66.74	24.05	—	34.03
Rush (receiver). (cylinders 24 in. and 38 in.; stroke 2¼ ft.)									
Steam-jacketed:—									
18. 1st cylinder	36.0	2.46	82.3	70.8	318.6	127.90	17.14	13.52	18.38
Both cylinders	—	6.22	—	—	—	266.50	—	—	—
19. 1st cylinder	59.8	1.60	50.2	55.5	249.7	63.70	19.74	16.94	22.10
Both cylinders	—	4.03	—	—	—	168.70	—	—	—

Table No. 117 (*continued*).

STEAMER: Condition of Cylinders. 1	Cut-off. 2	Actual Ratio of Expansion. 3	Total Maximum Pressure per square inch in 1st Cyl. 4	Turns per Minute. 5	Speed of Piston in feet per Minute. 6	Indicator Horse-power. 7	Weight of Steam per I.H.P. per Hour.		
							By Indicator 1st Cylinder. 8	By Indicator 2d Cylinder. 9	Actually Consumed. 10
	p. cent.	ratio.	lbs.	turns.	feet.	H.P.	pounds.	pounds.	pounds.
Dexter (single cylinder). (cylinder 26 in.; stroke 3 ft.)									
20. Felted and lagged...........	18.3	4.46	80.4	56.5	339.0	185.87	16.23	—	23.86
21. Do. do.	23.3	3.67	79.3	64.3	385.8	228.08	17.24	—	24.12
22. Do. do.	24.8	3.49	79.2	61.1	366.6	218.97	16.33	—	23.91
23. Do. do.	33.3	2.72	77.0	72.8	436.8	292.37	18.54	—	24.31
24. Do. do.	26.2	3.34	52.3	50.8	304.8	124.27	18.85	—	28.80
25. Do. do.	38.1	2.42	51.4	55.2	331.2	161.85	18.66	—	28.94
26. Do. do.	45.2	2.08	53.6	60.7	364.2	196.19	20.33	—	31.79
Dallas (single cylinder). (cylinder 36 in.; stroke 2½ ft.)									
27. Felted and lagged...........	13.3	5.07	46.9	48.7	243.5	137.96	19.20	—	26.69
28. Do. do.	19.7	3.89	47.0	56.9	284.5	186.80	20.72	—	26.96
29. Do. do.	26.4	3.13	46.6	61.5	307.5	221.45	20.06	—	26.94
30. Do. do.	28.8	2.94	45.9	64.5	322.5	242.81	21.20	—	28.90
31. Do. do.	38.6	2.32	39.4	63.5	317.5	234.30	22.76	—	30.99

In the Bache Woolf-receiver engine, having steam in the jacket of the second cylinder, the most efficient action of the steam is secured with an admission of from 32 per cent to 40 per cent, Nos. 6 and 7 in the table, expanding from six to seven times. In the Rush receiver-engine, likewise, an admission of 36 per cent, No. 18, expanding six and a quarter times, is more efficient than one of 60 per cent admission. Here is a consensus of evidence that an admission of, say 35 per cent, or say one-third of the stroke in the first cylinder, leading up to a total actual expansion of from six to seven times, is the most efficient for working the steam. The ratios of the cylinders of these two engines are about the same, being 2.440 and 2.509 respectively; but the cylinders of the Rush are larger and shorter in proportion to length than those of the Bache, whilst the maximum initial pressure in the Rush, 82.3 lbs. per square inch, is lower than that in the Bache, which is a mean of 92 lbs., at the same time that the back-pressures in the larger cylinders are identical. The principal points of distinction between the engines are that the second cylinder only of the Bache was steam-jacketed, whilst both cylinders of the Rush were jacketed, and the engine of the Rush was worked at a higher rate of speed than that of the Bache. Whilst it is a matter for doubt whether the relatively higher speed of the Rush engine materially influenced the result in its favour, the difference in the speeds may be set off against the difference of initial pressures; and it may be concluded that the employment of the steam jacket on the first cylinder as well as on the second cylinder of the Rush, against the absence of a steam jacket on the first cylinder of the Bache, led to the more efficient action of the steam in the Rush. In the Rush, 18.38 pounds of steam was consumed per indicator horse-power, against 20.35 pounds in the Bache.

In the Bache, No. 10, it is true, cutting off at 55.4 per cent in the first cylinder, and expanding 4.24 times, the consumption of steam per indicator horse-power was only 21.17 pounds against 22.10 pounds in the Rush, No. 19, cutting off at 59.8 per cent, and expanding 4.03 times. Here, the Bache has the advantage of cutting off a little earlier and expanding a little more than the Rush; and, accounting in this way for the better work of the steam in the Bache, it may be concluded that, under all the conditions, the engines were in this case on a footing of equality in point of efficiency of steam, and that the absence of a steam jacket from the first cylinder of the Bache did not operate materially against the engine. This conclusion follows independently from the conditions, since, as was before determined, the steam jacket is of much greater value for the shorter admissions than for the longer ones.

Working the second cylinder only of the Bache, No. 16, as a single cylinder, in cutting off at 16.3 per cent, and expanding 5.11 times, the steam used per horse-power is not much more than that used in the compounded cylinders, No. 9, with the same measure of expansion. That is to say, in cutting off at 45.2 per cent of the first cylinder, and expanding 5.10 times into the second cylinder, 22.38 pounds of steam are consumed per horse-power, against 23.15 pounds in the single second cylinder expanding 5.11 times; the maximum total initial pressure in both cases being about 90 lbs. per square inch. Plotting the results of Nos. 14, 15, 16, and 17 for the single second cylinder of the Bache, it appears that the consumption of water as steam per horse-power is practically the same for 20 per cent as for 16.3 per cent admission, with an expansion-ratio in the former case of 4.33.

Though the difference here is small, it is in favour of the compounded cylinders, and it quickly increases for greater ratios of expansion. Taking the mean of the first two cases, Nos. 4 and 5, of the compounded cylinder, expanding to 16.85 times and 9.19 times respectively, the mean admission is 16.6 per cent, the mean expansion-ratio is 13.02, and the mean consumption of steam is 22.91 pounds per indicator horse-power. With the single second cylinder, No. 14, on the contrary, cutting off at 4.2 per cent, and expanding 12.62 times, the consumption of steam is 27.11 pounds per horse-power, or 4.20 pounds more than in the compounded engine. Again, the compounded engine, No. 5, cutting off at 22.9 per cent, and expanding 9.19 times, consumed only 20.71 pounds of steam against 24.09 pounds in the single second cylinder, No. 15, cutting off at 8.1 per cent, and expanding 8.57 times.

From the evidence of these contrasts, it is clear that by the compounding of an engine in superadding a smaller cylinder to a given steam-jacketed cylinder, the efficiency of the steam is augmented, whether the smaller cylinder be steam-jacketed or not. The actual gains of efficiency may be here recapitulated:—

Gain by Compounding.

CYLINDERS.	Cut-off.	Expansions.	Steam used.	Economy.
	per cent.		lbs.	lbs.
Single, No. 14	4.2	12.62	27.11	4.20
Compound, mean of Nos. 4 and 5	16.6	13.02	22.91	
Single, No. 15	8.1	8.57	24.09	3.38
Compound, No. 5	22.9	9.19	20.71	
Single, No. 16	16.3	5.11	23.15	·77
Compound, No. 9	45.2	5.10	22.38	

The contrast of efficiencies between the work of steam in steam-jacketed cylinders, and the same cylinders deprived of steam in the jacket, is brought out in the following cases from the performance of the engine of the Bache:—

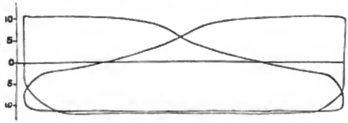
Steam in Jacket versus No Steam in Jacket.

COMPOUND CYLINDERS.	Cut-off.	Expansions.	Steam used.	Economy.
	per cent.		lbs.	lbs.
Steam not in jacket, No. 1	23.1	9.15	23.76	3.05
Steam in jacket, No. 5	22.9	9.19	20.71	
Steam not in jacket, No. 2	33.5	6.66	23.04	2.71
Steam in jacket, No. 6	31.8	6.98	20.33	
Steam not in jacket, No. 3	40.5	5.63	23.21	2.04
Steam in jacket, mean of Nos. 7 & 8	39.7	5.72	21.17	
SINGLE CYLINDERS.				
Steam not in jacket, No. 11	4.75	11.82	35.08	7.97
Steam in jacket, No. 14	4.2	12.62	27.11	
Steam not in jacket, No. 12	9.6	7.62	29.62	5.53
Steam in jacket, No. 15	8.1	8.57	24.09	
Steam not in jacket, No. 13	15.5	5.32	26.25	3.10
Steam in jacket, No. 16	16.3	5.11	23.15	

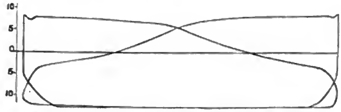
For the single cylinder, the use of the steam jacket makes a great increase of efficiency in reducing the consumption of steam per horse-power, as against the absence of steam from the jacket. There is also a gain of efficiency by the same means with the compound cylinders; and, in both cases, the gain increases as the ratio of expansion is increased. But the gain is much less for the compound cylinders than for the single cylinder—an unexpected comparative result which is due to the fact that only one of the compounded cylinders is jacketed, and also to the fact that the range of pressure and temperature is less in each of the compounded cylinders, between which it is distributed, and which are completely isolated from each other, than in the single cylinder.

The performances of the Dexter, not steam-jacketed, were made partly with higher maximum initial pressures, about 80 lbs. total, in Nos. 20 to 23; and partly with lower pressures, about 52 lbs. total, in Nos. 24 to 26. The first four, Nos. 20 to 23, are nearly alike in quantity of steam consumed per horse-power—about 24 pounds—although the periods of admission vary from 18.3 per cent to 33.3 per cent, and the expansions from 4½ to 2¾. In No. 24, at lower pressure, with an admission of 26.2 per cent, 28.8 pounds of steam are consumed per horse-power against, say, 24 lbs. of steam for 24.8 per cent admission at the higher pressure: making a material difference in favour of the higher pressure.

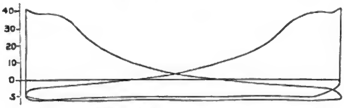
The Dexter shows, for equal admissions, lower consumptions of steam than the Dallas, which is also unjacketed. The most obvious difference of condition lies in the lower initial pressures in the Dallas than in the Dexter. There is also the difference of proportion of the cylinders, of which that of the Dallas has a length of stroke less than the diameter, whilst the stroke in the Dexter is greater than the diameter. The second cylinder of the Bache, also, has a stroke somewhat less than the diameter, agreeing in this respect with the Dallas. It is found, accordingly, that in working the second cylinder only of the Bache without steam in the jacket, the consumption of steam in No. 13 trial, for 15.5 per cent admission, is just a little less than that of the Dallas, No. 27, for 13.3 per cent of admission. That it should be less is in accordance with what is to be expected; and that the two consumptions should be so nearly alike, affords presumptive evidence that the consumption of steam has been affected to some extent by the proportions of the respective cylinders: that an unjacketed cylinder of which the stroke is less than the diameter, is less economical than one of a longer stroke.



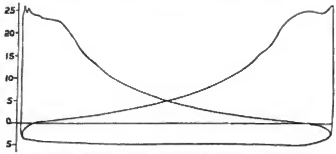
No. 3 Diagrams; 12½ lbs. pressure in boiler. Jacket in use.



No. 5 Diagrams; 12½ lbs. pressure in boiler. Jacket filled with 70-lb. steam.



No. 16 Diagrams; 40 lbs. pressure in boiler. Jacket in use.



No. 29 Diagrams; 70 lbs. pressure in boiler. Jacket in use.

Figs. 254.—Indicator Diagrams from the Gallatin.

The *single-cylinder condensing engine of the U.S. revenue steamer Gallatin*, of about the same capacity, was tested to compare with the compound engine of the Rush, by Chief-engineer Loring and Mr. Emery, in December and January, 1874–75.[1] There is one engine, inverted, with a thoroughly steam-jacketed and felted cylinder, 30.1 inches in diameter, with a stroke of 2½ feet. The total clearance at each end amounts to 6.60 per cent.

Table No. 118.—WORK OF EXPANDED STEAM:—AMERICAN MARINE CONDENSING ENGINE OF THE GALLATIN. December and January, 1874–75.

Effective Pressure in Boiler per square inch. Condition of Cylinder. (Cylinder, 34.1 in.; Stroke, 2½ ft.) 1	Cut-off. 2	Actual Ratio of Expansion. 3	Total Maximum Pressure per square inch. 4	Turns per Minute. 5	Speed of Piston in feet per Minute. 6	Indicator Horse-power. 7	Weight of Steam per I.H.P. per hour.	
							By Indicator. 8	Actually consumed. 9
About 12½ lbs.	per cent.	ratio.	lbs.	turns.	feet.	H.P.	pounds.	pounds.
1. Jacket not in use.........	46.3	2.01	25.3	40.05	200.2	86.98	25.62	40.37
2. Do.	64.0	1.51	23.7	40.9	204.5	90.17	29.71	44.17
3. Jacket in use...............	46.8	2.00	26.3	41.3	206.5	95.25	25.97	33.34
4. Do.	64.8	1.49	24.1	42.4	212.0	97.87	32.57	37.40
5. Do. filled with 70-lb. steam	52.5	1.80	23.9	41.1	205.5	89.65	26.88	34.06
6. Do. do.	62.6	1.54	24.9	42.3	211.5	102.91	28.95	34.86
About 40 lbs.								
7. Jacket not in use.........	11.4	5.92	56.3	43.0	215.0	122.84	17.89	26.01
8. Do.	13.9	5.21	52.8	44.2	221.0	127.20	18.49	26.70
9. Do.	22.0	3.73	55.8	50.8	254.0	182.23	18.28	24.00
10. Do.	27.1	3.16	51.8	50.1	250.5	182.36	19.12	26.32
11. Do.	32.6	2.72	52.3	56.0	280.0	263.87	19.48	24.49
12. Do.	41.3	2.23	50.3	55.8	279.0	236.46	22.21	28.10
13. Do.	69.1 throttled	1.41	44.7	52.7	263.5	219.46	26.11	29.59
14. Jacket in use...............	10.5	6.08	58.0	44.3	221.5	121.07	17.90	22.94
15. Do.	14.4	5.07	53.9	46.1	230.5	137.50	17.23	24.02
16. Do.	15.5	4.82	54.6	45.9	229.5	136.45	17.39	22.39
17. Do.	17.2	4.49	54.7	50.3	251.5	163.25	17.44	24.84
18. Do.	22.1	3.71	54.7	49.2	246.0	163.21	17.19	23.17
19. Do.	25.5	3.32	52.4	51.1	255.5	185.19	18.55	25.71
20. Do.	37.7	2.40	49.3	54.5	272.5	213.02	19.04	25.29
21. Do.	41.6	2.21	50.4	58.2	291.0	255.01	21.43	26.52
About 70 lbs.								
22. Jacket not in use.........	17.3	4.46	76.2	60.5	302.5	268.63	16.94	24.35
23. Do.	7.1	7.78	83.4	52.3	261.5	185.09	17.50	25.03
24. Do.	12.3	5.63	80.8	56.0	280.0	231.44	17.01	23.77
25. Do.	15.0	4.94	81.7	59.9	299.5	279.57	16.37	21.89
26. Do.	18.5	4.25	74.0	59.4	297.0	282.93	16.95	23.99
27. Jacket in use...............	17.1	4.50	77.8	61.5	307.5	281.65	16.68	22.04
28. Do.	8.0	7.32	85.0	51.1	255.5	196.99	15.67	20.49
29. Do.	12.2	5.68	85.3	58.6	293.0	270.39	16.72	20.61
30. Do.	14.8	4.98	83.5	61.2	306.0	306.24	16.88	20.69
31. Do.	17.3	4.46	81.7	61.3	306.5	295.57	18.14	21.42
32. Do.	18.9	4.19	80.7	68.7	243.5	347.98	17.18	21.48

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[1] *Journal of the Franklin Institute*, vol. lxxi. page 102; 1876.

Steam for the jackets is conducted through a felted pipe to the upper part of the cavity in the cylinder cover. This cavity is connected by a second pipe from the bottom of the cavity to the side jacket, which is one with the jacket of the lower cover, from which the condensation-water is blown into the hot-well, through a regulated valve. There is a short slide-valve, cutting off at two-thirds, and an expansion valve on the back of it. The pressure in the boiler was fixed successively at 12½ lbs., 40 lbs., and 70 lbs. effective pressure per square inch, for the purposes of the trials. The table, No. 118, contains the leading results of these trials. Sample indicator diagrams are shown in figs. 254, page 556.

To compare the consumption of steam per horse-power with different pressures in the boiler, and corresponding maximum initial pressures in the cylinder, when the cylinders are steam-jacketed:—

Steam Consumed for Different Pressures.

	Effective Pressure in Boiler.	Total Maximum Initial Pressure.	Per Cent Cut Off.	Expansions.	Pounds of Steam.
No. 3	12½ lbs.	24.3 lbs.	46.8	2	33.34
No. 21	40 "	50.4 "	41.6	2.21	26.52
					difference, 6.82
No. 14	40 lbs.	58 lbs.	10.5	6.08	22.94
No. 29	70 "	85.3 "	12.2	5.68	20.61
					difference, 2.33
No. 17	40 lbs.	54.7 lbs.	17.2	4.49	24.84
No. 31	70 "	81.7 "	17.3	4.46	21.42
					difference, 3.42

Here it is shown that the steam is worked with a greater degree of efficiency at the comparatively higher pressures.

The influence of the higher pressure of steam in the jacket,—70 lbs. effective, in place of 12½ lbs. steam from the boiler, is shown, in comparing Nos. 4 and 6 trials, to lead to the reduction of the steam per horse-power per hour from 37.40 pounds to 34.86 pounds, in cutting off at 64.8 per cent and 62.6 per cent; making a difference of 2.54 pounds, or 6¾ per cent.

The most effective admissions appear to be 15½ per cent with 4.82 expansions, for 40 lbs. steam; and from 8 per cent to 15 per cent, with 7.32 to 5 expansions for 70 lbs. steam, consuming 20½ lbs. of steam per horse-power per hour.

This result may be compared with the best performance of the Rush, No. 18 in table No. 117, where it appears that for 6¼ expansions, in cutting off at 36 per cent in the first cylinder, with 82.3 lbs. initial pressure, 18.38 lbs. of steam are consumed per horse-power per hour. The difference in favour of the compound engine of the Rush is more than 2 pounds of steam, or 10.3 per cent. This deduction is the more conclusive since the proportions and capacities of the cylinders are nearly alike. The stroke in each case is less than the diameter, and the net capacities are 7.10 cubic feet for the Gallatin, and 7.13 cubic feet for the second cylinder of the Rush. The average back-pressure in the Gallatin was 3.92 lbs. total pressure against 3.44 lbs. in the second cylinder of the Rush.

When the steam was not admitted into the jackets, the engine of the Gallatin consumed a greater quantity of steam than when steam was admitted. With 12½ lbs. steam in the boiler, the use of the steam-jacket led to a reduction of about 7 pounds of steam per horse-power, or about 17 per cent of the steam consumed, when there was no steam in the jacket. For the higher pressure, 40 lbs. in the boiler, the consumptions of steam were from 26 pounds to 28 pounds without steam in the jacket; and from 23 pounds to 26½ pounds with steam, making a reduction of from 3 pounds to 1½ pounds, or from 11½ per cent for the greater expansions to 5¼ per cent for the less expansions. For 70 lbs. steam in the boiler, a reduction of from 4½ pounds to 2½ pounds was effected, or from 14 per cent to 10 per cent for the greater and the less expansions respectively. The lowest rate of consumption without steam in the jacket (No. 25), is 21.89 pounds of water from the boiler, cutting off at 15 per cent, with 4.94 expansions, and 81.7 lbs. total maximum pressure per square inch in the cylinder.

The *compound engines of the French naval steamer Cher* were subjected to experiment at Cherbourg in 1866. The results of the experiments were investigated by Mr. Isherwood.[1] The engines have three horizontal cylinders of equal diameters, 43.30 inches, and equal strokes, 1.64 feet, connected to one crank-shaft:—the two outer cylinders by cranks at 90°, and the middle cylinder by a crank at 135° with each of the two outer cranks. The ratio of the first to the two others, in net capacity, is 2. The steam from the boiler is cut off at 88 per cent of the stroke of the middle cylinder; whence it is expanded, through the receiver, into the two outer cylinders simultaneously, in each of which it is cut off at 78 per cent of the stroke. From these, the steam is exhausted into surface-condensers. The admission to the middle cylinder could be varied by means of an expansion-valve. The steam is not cushioned or compressed in any cylinder. The cylinders are not steam-jacketed; but the steam is superheated. The experiments were made, 1st, with a constant pressure in the boiler, and various degrees of expansion, with free admission of steam; 2d, with a lower constant pressure in the boiler, higher degrees of superheat, and various degrees of expansion; 3d, with steam of different pressures in the boiler, and the same degree of expansion, without throttling; 4th, with different degrees of throttling; 5th, with saturated steam admitted to the end cylinders only, worked as single cylinders. Selected results of the trials, excepting those for throttling, which have no significance, are given in table No. 119. Of these Nos. 1 and 5 are identical with Nos. 9 and 11 respectively.

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[1] See *Minutes of Proceedings of the Institution of Civil Engineers*, vol. liv. page 335. Also *Journal of the Franklin Institute*, July, August, 1878; pages 1, 73.

Table No. 119.—ECONOMY OF SUPERHEATED STEAM IN THE WAR STEAMER CHER.

CONDITIONS. (Cylinders 43.3 inches: Stroke 1.64 feet.) 1	Cut-off in 1st Cylinder. 2	Actual Ratio of Expansion 3	Effective Pressure in the Boiler, per sq. inch. 4	Absolute Pressure in Steam-pipe near Cylinder, per sq. in. 5	Superheat of Steam on entering Middle Cylinder. 6	Absolute Pressure in Receiver, per sq. in. 7	Turns per Minute. 8	Speed of Piston in feet per Minute. 9	Indicator Horse-power.			Weight of Steam per Indicator Horse-power per Hour.	
									1st Cyl.	2d Cyl. 10	Total.	At Cut-off in 1st Cyl. 11	Actually Consumed 12
I.	per cent.	ratio.	lbs.	lbs.	degrees Fahr.	lbs.	turns.	feet.	H.P.	H.P.	H.P.	pounds.	pounds.
1. Varied expansion: maximum pressure...	88	2.26	27.3	41.2	11.5	34.3	66.3	217.5	182.3	309.8	492.1	21.4	23.5
2. Do. do. ...	60	3.21	27.7	41.7	16.2	29.1	61.4	201.4	171.1	199.4	370.5	18.6	22.1
3. Do. do. ...	50	3.78	27.3	41.6	10.8	27.2	58.1	190.6	156.8	162.2	319.0	17.3	22.0
4. Do. do. ...	40	4.59	27.9	42.0	18.0	25.5	54.6	179.1	148.4	121.4	269.8	16.3	22.4
5. Varied expansion: medium pressure...	88	2.26	19.6	33.6	22.1	30.1	58.8	192.9	126.9	210.6	337.5	22.7	25.1
6. Do. do. ...	60	3.21	19.5	33.5	20.8	25.5	53.7	176.2	126.5	127.6	254.1	18.6	22.9
7. Do. do. ...	50	3.78	19.4	33.4	22.7	24.5	51.1	167.6	114.0	109.1	223.1	17.7	23.3
8. Do. do. ...	40	4.59	19.7	34.1	21.5	23.7	48.1	157.8	98.1	91.0	189.1	16.3	23.7
II.													
9. Constant expansion: various pressures...	88	2.26	27.3	41.2	11.5	34.3	66.3	217.5	182.3	309.8	492.1	21.4	23.5
10. Do. do. ..	88	2.26	21.1	35.0	21.7	31.3	61.3	201.1	140.5	236.2	376.7	22.7	25.0
11. Do. do. ...	88	2.26	19.6	33.6	22.1	30.1	58.8	192.9	126.9	210.6	337.5	22.7	25.1
12. Do. do. ...	88	2.26	12.2	26.6	30.3	26.8	51.3	168.3	82.5	142.0	224.5	23.3	26.4
III.													
13. Saturated steam: 2d cylinders as single cylinders.........	78	1.26	18.5	32.8	12.5	30.0	66.3	217.5	—	491.6	491.6	—	33.7

By the table it appears that the most economical expansion-ratio in the engines of the Cher, for both the maximum and the medium boiler pressures, is about 3½, the steam being cut off at 55 per cent in the first cylinder. These two values are means of Nos. 2 and 3, and Nos. 6 and 7; but the economy is nearly the same for expansion-ratios of 3.21 and 3.78, and it may be taken that the maximum economy belongs to expansion-ratios of from 3¼ to 3¾. The steam was not sufficiently superheated to prevent its passing into the second cylinder in the saturated state. On the contrary, the temperature of the steam in the receiver, between the first and middle cylinder, and the second or outer cylinders, was found to be that due to the observed pressure for saturated steam. The steam had, therefore, when it reached the receiver, parted with the superheat, and had become, in fact, saturated. But this evidence is not needed to prove the fact of an early reduction of the steam to a state of saturation, seeing that while yet the steam was in the first cylinder, the weight of sensible steam at the point of cut-off, column 11, taken as saturated, was considerably less than the weight actually consumed, column 12: a proof of initial condensation.

The evidence of the table, so far as it goes, clearly shows that the higher the working pressure, the more economical is the performance. In Nos. 9 to 12 trials, this relation is proved, even though the steam is at the same time superheated to a higher degree for the lower pressures.

In the trial, No. 13, made with saturated steam in the second cylinders, as single, the consumption of steam, 33.7 pounds per horse-power, is far in excess of the consumption in the compounded cylinders of the same engine, which, in the case of No. 5, with equal boiler-pressure, though with greater expansion, is only 25.1 pounds.

Several *Woolf beam pumping engines* were tested by Mr. J. G. Mair, who, in his paper on "The Independent Testing of Steam Engines,"[1] publishes the results of several trials, conducted by him, of these compound engines, for which the averaged indicator diagrams are given in figs. 255, 256. He exhaustively analysed the results of these trials. Some of the results are given in table No. 120, following.

Trials B. C.—A Woolf beam rotative engine, made by Simpson & Co., working a deep-well pump direct from the beam. The cylinders are 22 inches and 34 inches in diameter respectively, with strokes of 3 feet 7 inches and 5½ feet; and are thoroughly steam-jacketed. There is a long slide with the equilibrium passage in it, and expansion-plates on the back, worked by excentrics.

Trials D, E, F.—A Woolf beam engine, driving a flour mill, made by Simpson & Co. The cylinders are 24½ inches and 38 inches in diameter respectively, with strokes of 3 feet 5 inches and 5½ feet; steam-jacketed on the sides only. The slide with cut-off plates on the back, for the first cylinder, and a piston-valve for the second cylinder, are worked by excentrics.

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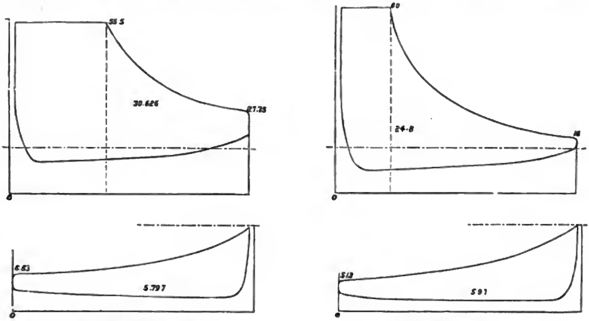
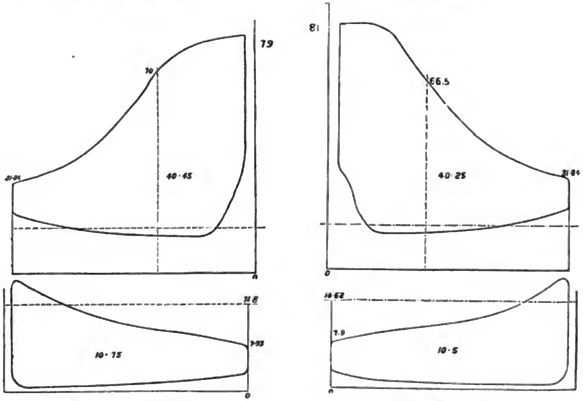
[1] See foot-note, page 522.

Trials G, H.—A horizontal Woolf tandem engine, not jacketed, made by Pollit and Wigzell, having cylinders 15¾ inches and 28½ inches in diameter respectively, with strokes of 4 feet 3 inches. There is a single slide on the first cylinder, with adjustable cut-off plates on the back. The load on the engine was light when the two trials recorded were made, and the steam was much wiredrawn.



Woolf Beam Engine, B.

Same Engine, C.

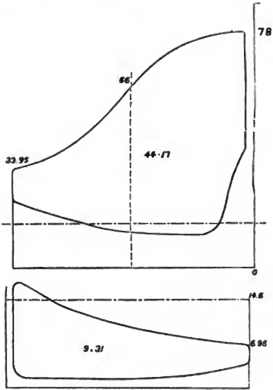


Woolf Beam Engine, D.

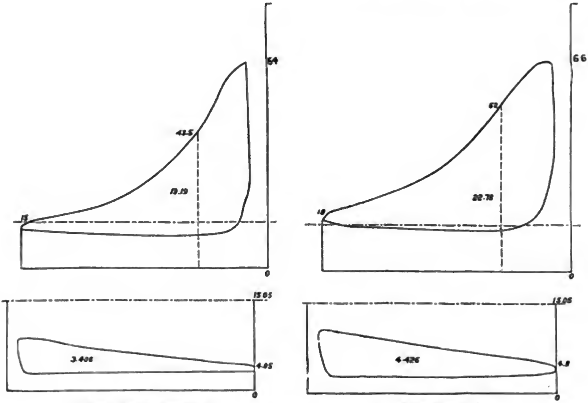
Same Engine, E.

Figs. 255.—Indicator Diagrams from Woolf Engines. J. G Mair.

Trial J.—The results of this trial, already discussed at page 535, with indicator diagrams, are repeated in the table as given by Mr. Mair. There are one or two slight differences between the two records, arising from the fact that Mr. Mair brought into the calculation the space occupied by the piston-rod, which was not allowed for in the author's discussion of the results. The engine is a compound beam receiver-engine. The cylinders are 21 inches and 36 inches in diameter respectively, with a stroke of 5½ feet. The cylinders and the receiver, on Cowper's system, are steam-jacketed, except the covers of the first cylinder and the receiver. The cranks are at right angles, and two low-lift plunger pumps are driven direct from each of the beams. There are two slide-valves, one at each end of the cylinder, with adjustable expansion-plates. Two trials were made, exactly alike, of which the results of one are given.



Woolf Beam Engine, F (same as D, E).



Woolf Tandem Engine, G.

Same Engine, H.

Figs. 256.—Indicator Diagrams from Woolf Engines. J. G. Mair.

In the table No. 120, particulars of the engines are given in columns 1 to 6; the state of the jacketing in column 7; the net pressure in the boiler in column 8; the cut-off, or period of admission in the first cylinder, in column 9; the actual working ratio of expansion in column 10; the total actual ratio of expansion in column 11. The net actual or working ratio of expansion, column 10, is deduced from data given in the paper, it being here assumed that the free intermediate expansion in the Woolf and tandem engines was in the ratio of 1 to 1.20. The total expansion, column 11, has been divided by 1.20 to find the total working expansion, in column 10. The total maximum pressure in the cylinder, column 12, and the total initial pressure for expansion at the cut-off, column 13, have been measured from the indicator diagrams. Columns 14, 15, and 16 are obvious. Column 17 contains the weight of dry saturated steam per indicator horse-power, and the contents of column 18 are calculated from them according to the given percentages of priming.

The lowest consumption of steam is exhibited by the receiver-engine J. It surpasses all the Woolf engines in this respect. Of these, two are but partially steam-jacketed; but the first engine B, C, is, like J, thoroughly jacketed, and cuts off at the same point of the stroke, 21 per cent; yet it consumes 17.34 pounds of steam per indicator horse-power, against 14.84 pounds by engine J, the employment of the receiver in which, combined with the steam-jacketing of it, is most probably the cause of the greater efficiency. Comparison of Woolf engines C, and D, E, is complicated by wiredrawing in D, E, which, in connection with higher maximum pressures, aids in procuring nearly the same degree of efficiency in trials D, E as in trial C, although the ratio of expansion in the first cylinder of D, E is much less than in C; and although, also, the cylinders are not completely jacketed in D, E, a greater quantity of steam was used in trials G, H, with jacketless cylinders.

The results of the trials B and F, without steam in the jackets, show, as usual, higher rates of consumption of steam than when there is steam in the jackets, as follows:—

C, Woolf—steam in jackets	17.73	lbs. per I.H.P.
B, Do. —no steam in jackets	28.88	„ „
Excess	11.15	
E, Woolf—steam in jackets (sides)	17.83	„ „
F, Do. —no steam in jackets	19.72	„ „
Excess	1.89	

There has evidently existed an exceptional cause of the large excess of steam used in the B trial. One reason for less steam having been used in trial F than in trial B, is that the cylinder-pressures in trial F were the higher. For the same reason, apparently, a greater quantity of steam was used in trials G and H, with jacketless cylinders.

Table No. 120.—RESULTS OF TESTS OF COMPOUND ENGINES. MR. J. G. MAIR, 1882.

ENGINE AND TEST-MARK.	1st Cylinder.		2d Cylinder.		Capacity-ratio of Cylinders.	Steam-jacketing.	Net Pressure in Boiler per Square Inch.	Cut-off in 1st Cylinder.	Actual Working Ratio of Expansion.	Total Actual Ratio of Expansion.	Total Maximum Pressure in Cylinder per Square Inch.	Total Initial Pressure at Cut-off per Square Inch.	Turns per Minute.	Speed of Piston in Feet per Minute.	Indicator Horse-power.	Dry Saturated Steam per I.H.P. per Hour.	Water as Steam from the Boiler per I.H.P. per Hour.
	Dia-meter.	Stroke	Dia-meter.	Stroke													
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
	inches	inches.	inches.	inches.	ratio.		lbs.	p.cent.	ratio.	ratio.	lbs.	lbs.	turns.	feet.	I.H.P.	pounds.	pounds.
B. Woolf, beam..	22	43	34	66	3.66	thoroughly	43	39.2	1st cyl. 2.524 2d ,, 3.071 7.750	9.30	55.5	55.5	17.8	196.2	75.9	26.62	28.88
C. Do. do. ..	,,	,,	,,	,,	,,	,,	47	21	1st cyl. 4.280 2d ,, 3.071 13.133	15.76	60	60	19.6	215.8	75.2	17.34	17.73
D. Woolf, beam..	24½	41	38	66	3.87	at the sides	70	36.6	1st cyl. 2.517 2d ,, 3.200 8.033	9.64	79	70	33.7	371.0	267.9	17.73	18.17
E. Do. do. ..	,,	,,	,,	,,	,,	,,	73	37	1st cyl. 2.485 2d ,, 3.200 7.967	9.56	81	66.5	34.2	376.4	267.9	17.39	17.83
F. Do. do. ..	,,	,,	,,	,,	,,	,,	74	49.2	1st cyl. 2.031 2d ,, 3.200 6.481	7.77	78	66	34.5	379.7	267.9	19.24	19.72
G. Horizontal Woolf, tandem	15¾	51	28½	51	3.27	no jacket	71	23.3	1st cyl. 3.526 2d ,, 2.715 9.571	11.48	64	43.5	80.5	683.8	120.3	21.65	22.52
H. Do. do.	,,	,,	,,	,,	,,	,,	70	22.2	1st cyl. 3.574 2d ,, 2.715 9.704	11.64	66	52	81.5	692.8	149.7	20.88	21.71
J. Receiver, beam	21	66	36	66	2.94	thoroughly, with receiver	61	21	1st cyl. 4.491 2d ,, 2.233 10.031	13.61	75	64	24	263.8	127.40	14.84	15.19

CHAPTER X.

GENERAL DEDUCTIONS FROM THE FOREGOING DATA OF THE PERFORMANCE OF STEAM ENGINES.

I. STEAM BROUGHT INTO EVIDENCE BY THE INDICATOR DIAGRAM.

In summarizing the discussions in the last two chapters, the first deduction of importance is the fact that in certain instances of thoroughly steam-jacketed cylinders, the whole of the steam passed into the cylinder has been brought into evidence by the medium of the indicator diagram:—namely, the steam that is admitted, expanded, and passed right through into the exhaust passage, the steam that is condensed and re-evaporated, and the proportion of steam of which the heat is liberated by condensation during expansion and converted into work. Thus is effectually exploded the hypothesis entertained by Zeuner and others that there is a quantity of water constantly in the cylinder which alternately receives heat from the steam admitted, and evolves it. In the case of the use of superheated steam, the superheat, also, is brought into evidence, in two modes:—by the conversion of this heat into work, and by the adiabatic character of the expansion-curve of pressure, for which the equation is given at page 495, as that of the adiabatic expansion of gaseous steam. The adiabatic curve, and thence the evidence of superheat in the steam, has been effectively exemplified in the indicator diagrams of the Great Britain and other inside-cylinder locomotives, as discussed in pages 495 and 496, and shown in table No. 109

In the investigation of the performance of the Porter-Allen engine using superheated steam, pages 502 and 503, it has been shown that the work-heat of expansion, for one stroke of the piston, 12.19 units, was contributed to the extent of 2.10 units by superheat; 3.68 units by constituent heat set free, being the difference of the constituent heats of the two quantities of steam at the initial and the final pressures of expansion; and 6.41 units by the condensation of expanding steam. The steam thus condensed, to provide 6.41 units of heat for one stroke, weighs .0066 pound, and, added to .1240 pound, the sensible expanded steam, makes up .1306 pound of steam brought into evidence by the indicator; or, after deduction of steam compressed, within 1.44 per cent, by excess, of the measured weight of water as steam actually consumed.

Applying a like calculation to the results of the performance of the pumping engines of the East London Water-works, table No. 113, page 512, there is a considerable balance of water as steam actually consumed, not brought into evidence by the indicator. The discrepancy has already, page 514, been ascribed to the incomplete jacketing of the cylinders, and the deficient recuperative action of the cylinder in re-evaporating condensed steam.

A like deficit takes place in the calculations for the three Corliss and Wheelock engines, page 515, in which, when condensing, about 81 per cent only has been brought into evidence by the indicator, and an average of about 89 per cent, when non-condensing—resulting from the absence of steam-jackets.

The work of the condensing engine of Haussmann & Co., using superheated steam, pages 520 and 521, affords instructive contrasts. When the steam had been sufficiently superheated, the whole of the steam actually consumed was, substantially, brought into evidence by the indicator; but, when the steam was saturated, the cylinder not having been steam-jacketed, there were large excesses of unrestored steam—steam not brought into evidence by the indicator.

Turning to the compound engines, in the condensing Woolf beam engine of Bryan Donkin & Co., the net steam consumed in the cylinder, brought into evidence by the indicator, calculated from a single set of diagrams, page 527, was at the rate of 20.30 pounds per indicator horse-power per hour, whilst the average measured consumption was only 18.50 pounds. The difference by excess is, as it happens, on the side of the indicator, incidental to the fact that the diagrams are but a single set. The whole of the steam consumed is thus put in evidence, with a margin, and for an actual expansion-ratio ranging up to 9.276 times, or a total expansion-ratio of 11.592.

A like excess of evidence by the indicator, in a single set of diagrams, was observed in the case of the Hele compound engine, made by Bryan Donkin & Co., page 531, the indicator showing a consumption within the cylinder of 21.67 pounds of steam per indicator horse-power per hour, against 18.76 pounds as measured. The discrepancy is due, no doubt, as before, to the incidental deviation of the special conditions under which the diagrams were taken, from average conditions. Otherwise, of course, they would have proved too much.

In the Woolf pumping engines, by Simpson & Co., at Thames Ditton, page 533, the steam-jackets were ineffective; and the cylinders acted almost as unjacketed cylinders. Accordingly, the indicator brought into evidence only 17.15 pounds of steam per indicator horse-power per hour, out of 22.62 pounds consumed from the boiler, in addition to what passed through the jacket. Here is a large deficit,—5.43 pounds, or 24 per cent of the steam passed into the cylinder permanently condensed.

The receiver compound beam engines, by Simpson & Co., at Thames Ditton, page 535, steam-jacketed, were worked with the jackets in full play; and the quantity of steam brought into evidence by the indicator, as having passed into the cylinder, amounted to 13.01 pounds per indicator horse-power per hour, against 13.17 pounds of dry steam actually delivered from the boiler, and passed through the cylinders, differing by little over 1 per cent. In this case, the evidence is of the greater value, as the diagram given is the carefully drawn average of a great number of diagrams taken in the course of the test.

A similar close coincidence between the calculated and measured quantities of water as steam passed into the cylinders, is proved in the case of the compound beam pumping engines, by Mr. Leavitt, at Lawrence, Massachusetts, page 540:—13.91 pounds against 14.06 pounds, including jacket-steam, per indicator horse-power per hour,—within 1 per cent of each other.

It is a significant fact that the whole of the steam passed into the cylinder, as deduced from actual measurement of feed-water, can be brought into evidence by means of the indicator diagram, in the manner exemplified in the several instances above cited. It is demonstrated that, by the instrumentality of the steam-jacket, or by direct superheating of the steam, heat can be supplied in quantity sufficient to prevent, or at least to mitigate, initial condensation in the cylinder, and to re-evaporate the whole of such steam as may thus be condensed, independently of what is condensed in yielding work-heat during expansion. Where, on the contrary, the steam-jacketing is insufficient, or is not applied at all, and the steam is delivered into the cylinder in the saturated state, there is, for considerable degrees of expansion, a large permanent deficit of steam, the quantity of steam brought into evidence by means of the indicator being, to an important extent, less than that which is delivered into the cylinder. Here is a contrast which goes to prove that the large deficits occasionally observed are not necessarily caused by leakage of steam at the pistons or the valves, since the same cause would, if real, effect the same deficits of superheated or steam-jacketed steam,—deficits which, as a matter of fact, it has been shown do not take place.

II. ECONOMY OF EXPANSIVE WORKING UNDER VARIOUS CONDITIONS:— SINGLE CYLINDERS.

Single Cylinders with Superheated Steam, Non-condensing.—It has been shown, page 496, that in the Great Britain and other inside-cylinder locomotives in which the cylinders and the steam-pipes were suspended in and enveloped by the hot gases in the smoke-box, the cylinders became highly heated, and the steam from the boiler became superheated, and as such expanded adiabatically in the cylinders.

In the Porter-Allen non-condensing engine, page 501, in which the steam was superheated to the extent of 35°.5 F., the expansion partook of the adiabatic character, alternate condensation and re-evaporation having been much less active than they would have been if saturated steam had been delivered into the cylinder. The comparatively great consumption of water as steam in the Porter-Allen engine, cutting off at 11 per cent, shows that this was not the most economical point of cut-off.

In M. Hirn's engine, page 520, cutting off at 28.7 per cent, non-condensing, the net pressure at cut-off was only 34 lbs. per square inch, and 30.65 pounds of steam superheated 130° F., was consumed per indicator horse-power per hour. The high rate of consumption arises from the low net pressure, above the atmosphere, together with a low ratio of expansion.

The results from the Great Britain—that is, the water consumed per indicator horse-power, page 497—may be taken for a standard of economical performance for non-condensing engines, the pressure being high, and the steam being sufficiently superheated to obviate alternate condensation and re-evaporation in the cylinder. Plotting the results to a base-line for the stroke, with ordinates for the weights of steam consumed for the several periods of admission set off on the base-line; and tracing a medium curve through the upper ends of the ordinates, the following is the equation of the curve:—

$$w = 16 + .10a + .0014a^2 \ldots\ldots\ldots\ldots \quad (1)$$

a = Period of admission, as a percentage of the stroke.
w = Water as steam consumed per horse-power per hour, in pounds.

The table, No. 121, contains a selection of data measured off the plotting curve, for which the equation (1) may be taken as representative of the efficiency of superheated steam in well-protected cylinders, non-condensing, worked with the ordinary slide-valve, with steam of 100 lbs. effective pressure per square inch in the boiler, and, say, a maximum effective pressure of 80 lbs. per square inch in the cylinder.

Table No. 121.—SINGLE CYLINDERS, WELL PROTECTED AND HEATED, NON-CONDENSING:—CONSUMPTION OF SUPERHEATED STEAM.

Net Pressure in Boiler, 100 lbs. per square inch; net maximum Pressure in Cylinders, 80 lbs. per square inch.

GREAT BRITAIN LOCOMOTIVE.

Period of Admission or Cut-off.	Actual Ratio of Expansion, reckoned to the End of the Stroke.	Water as Steam consumed per Indicator Horse-power per hour.
per cent.	ratio.	pounds.
20, or one-fifth	3.91	18.5
25, or one-fourth	3.31	19.4
30	2.87	20
33, or one-third	2.65	20.5
35	2.53	21.2
40, or two-fifths	2.26	22.25
50, or one-half	1.86	24.5
60	1.59	27
67, or two-thirds	1.44	29
70	1.39	30
75, or three-fourths	1.30	31.5
80, or four-fifths	1.23	33

Single Cylinders, not Steam-jacketed, Non-condensing.—The plotted results from the locomotive engines, Nos. 42 and 48, C.R., yield the consumption of water as steam given in table No. 122. They are not directly comparable with those of the Great Britain, unless allowance is made for the difference of ruling pressures.

The average results of the three American engines, Corliss and Wheeler, when non-condensing, page 515, show that for an admission of 15.5 per cent, expanding 5.81 times, 23.79 pounds of water as steam was consumed per horse-power per hour, with 104 lbs. net maximum pressure in the cylinder, and 29.36 lbs. per square inch, effective mean pressure. This, as an economical result, does not compare favourably with those in the table, considering the much higher pressures employed in the American engines; and it is clear that the admission, 15.5 per cent, is much below the admission for securing maximum economy.

Table No. 122.—SINGLE CYLINDERS, UNJACKETED, NON-CONDENSING:—
CONSUMPTION OF STEAM.

Net Pressure in Boiler, 80 lbs. per square inch; net maximum Pressure in Cylinder, about 60 lbs. per square inch.

C.R., NOS. 42 AND 48, LOCOMOTIVES.

Period of Admission or Cut-off.	Actual Ratio of Expansion, reckoned to the End of the Stroke.	Water as Steam consumed per Indicator Horse-power per hour.
per cent.	ratio.	pounds.
35, or about one-third..........	2.62	26
40, or two-fifths..................	2.33	27
50, or one-half..................	1.91	29.5
60	1.62	32
67, or two-thirds.................	1.46	34.5

Single Cylinders, Steam-jacketed, Non-condensing.—The results from portable engines, page 499, show what can be done for small cylinders by steam-jacketing them. But the scope of the observations for high maximum pressures in the cylinder, of from 72 lbs. to 77 lbs. per square inch, is limited to 32 per cent of admission. The mean of the results of Nos. 11 and 15 engines, may be taken as the most economical—showing an admission of 20 per cent, or one-fifth, and a consumption of 25 pounds of water as steam per indicator horse-power per hour. The consumptions of water as steam for lower maximum pressures are much higher than this. These results are to be taken as for cylinders of small size—8 or 9 inches in diameter.

Single Cylinders with Superheated Steam, Condensing.—The results from M. Hirn's engine, page 520, and the engines of the Cher, page 560, in both of which superheated steam was used, are as follows:—

ENGINE.	Cylinder, Diameter and Stroke.	Coating.	Super-heat.	Cut-off.	Actual Expansion-ratio.	Total Maximum Pressure in Cylinder, per sq. inch.	Total Cut-off Pressure, per sq. inch.	Water as Steam per Indicator Horse-power per hour.
	ins. ins.		degrees. Fahr.	per cent.	ratio.	lbs.	lbs.	pounds.
Hirn, No. 5	23¾ × 67	lagged	85	16.3	5.84	66.64	55.64	16.16
Do. No. 3	"	"	121	21.4	4.51	63.28	58.88	15.69
Do. No. 1 (corrected).	"	"	150	25.7	3.75	69.35	60.35	15.63
Do. No. 4	"	"	135	45.4	2.18	64.79	32.79	18.33
Do. No. 7	"	"	130	45.4	2.18	37.32	24.82	18.40
Cher........	43.3 × 19.7	lagged	12.5	78.0	1.26	32.80	—	33.70

The high consumption in the Cher shows that the small extent of superheat, 12½ degrees, had not much influence in economizing steam.

It was due, in part, to the lowness of the maximum pressure; and, no doubt, also, to the extreme shortness of the stroke of the cylinder. But the superheating of steam in Hirn's engine, 150 degrees, in cutting off at 25.7 per cent, has limited the consumption to 15.63 pounds per horse-power per hour. In cutting off at 21.4 per cent, and superheating only 121 degrees, an equal consumption, 15.69 pounds, was attained. The most economical period of admission is from 20 per cent to 25 per cent, with an actual ratio of expansion of from 4½ to 3¾. For this admission, with a maximum pressure of, say, 65 lbs. per square inch, and with 150 degrees of superheat, the consumption is, say, 15½ pounds of steam per indicator horse-power per hour.

Single Cylinders, not Steam-jacketed, or with Steamless Jackets, Condensing.—The engines of Corliss and Wheeler, page 516; Hirn, page 520; the Bache, page 552; the Dexter, the Dallas, and the Gallatin, pages 553 and 557, supply instances of performance of condensing engines, which are either not provided with steam-jackets, or are worked for the occasion without steam in the jackets. The results are collected in the following tablet:—

Single Cylinders, not Steam-jacketed, Condensing.

ENGINE.	Cylinder, Diameter and Stroke.	Cut-off.	Actual Expansion-ratio.	Total Maximum Pressure in Cylinder per square inch.	Total Cut-off Pressure per sq. inch.	Water as Steam per Indicator Horse-power per hour.
	ins. ins.	per cent.	ratio.	lbs.	lbs.	pounds.
Corliss & Wheeler, average..........	18 × 48	12.5	6.95	104.4	98.4	19.58
Hirn, No. 6............	23¾ × 67	16.3	5.84	61.5	54.5	19.93
Do. No. 2 (corrected)	,,	25.7	3.75	59.0	53.7	22.41
Mair, M................	32 × 66	24.6	3.84	54.5	45.7	26.46
Bache, No. 11.........	25 × 24	4.75	11.82	89.3	—	35.08
Do. No. 12.........	,,	9.6	7.62	91.1	—	29.62
Do. No. 13.........	,,	15.5	5.32	87.7	—	26.25
Dexter, No. 20.........	26 × 36	18.3	4.46	80.4	—	23.86
Do. No. 21.........	,,	23.3	3.67	79.3	—	24.12
Do. No. 22.........	,,	24.8	3.49	79.2	—	23.91
Do. No. 24.........	,,	26.2	3.34	52.3	—	28.80
Do. No. 23.........	,,	33.3	2.72	77.0	—	24.31
Do. No. 25.........	,,	38.1	2.42	51.4	—	28.94
Do. No. 26.........	,,	45.2	2.08	53.6	—	31.79
Dallas, No. 27.........	36 × 30	13.3	5.07	46.9	—	26.69
Do. No. 28.........	,,	19.7	3.89	47.0	—	26.96
Do. No. 29.........	,,	26.4	3.13	46.6	—	26.94
Do. No. 30.........	,,	28.8	2.94	45.9	—	28.90
Do. No. 31.........	,,	38.6	2.32	39.4	—	30.99
Gallatin, No. 23......	30.1 × 30	7.1	7.78	83.4	—	25.03
Do. No. 24......	,,	12.3	5.63	80.8	—	23.77
Do. No. 25......	,,	15.0	4.94	81.7	—	21.89
Do. No. 22......	,,	17.3	4.46	76.2	—	24.35
Do. No. 26......	,,	18.5	4.25	74.0	—	23.99

The comparatively low maximum pressures in the Dallas lead to greater consumptions of steam than in the Dexter or the Gallatin. The Dexter itself supplies contrasts of this kind. Thus, for Nos. 22 and 24 of the Dexter, cutting off at about 25 per cent, having about 80 lbs. and 52 lbs. total maximum pressures respectively, No. 24 consumed 5 lbs. more steam than No. 22.

The ranges of admission in the short-stroke cylinders for the greatest efficiencies are considerable. The Dexter, for admissions ranging from 18.3 per cent to 33.3 per cent, uses practically equal quantities of steam per horse-power. So does the Dallas, ranging from 13.3 per cent to 26.4 per cent admission. In the Gallatin, for admissions ranging from 12.3 per cent to 18.5 per cent, the consumption of steam fluctuates from 23.77 pounds to 21.89 pounds and 23.99 pounds.

The long-stroke cylinders of Corliss & Wheeler, and Hirn, show nearly equal consumptions of steam, 19.58 and 19.93 pounds respectively, cutting off at 12.5 per cent and 16.3 per cent. These results are here summarized:—

Single Cylinders, not Steam-jacketed, Condensing:—Summarized Results.

ENGINE.	Cylinder, Diameter and Stroke.	Cut-off.	Total Maximum Pressure per square inch.	Water as Steam per Indicator Horse-power per hour.
LONG-STROKE.	ins. ins.	per cent.	lbs.	pounds.
Corliss & Wheeler (average)...	18 × 48	12.5	104.4	19.58
Hirn	23¾ × 67	16.3	61.5	19.93
SHORT-STROKE.				
Bache	25 × 24	15.5	87.7	26.25
Dexter, Nos. 20, 21, 22, 23....	26 × 36	18.3 to 33.3% average 25%	79.0	24.05
Dallas, Nos. 27, 28, 29.........	36 × 30	13.3 to 26.4% average 19.8%	46.8	26.86
Gallatin, Nos. 24, 25, 22, 26...	30.1 × 30	12.3 to 18.5% average 15.8%	78.2	23.50

If in the Hirn cylinder steam of as high a maximum pressure had been used as in the Corliss & Wheeler cylinders it would probably have worked with as low a consumption as these; and, further, 16.3 per cent admission in the Hirn cylinder is decidedly more efficient than 25.7 per cent in the same cylinder, and than 24.6 per cent in Mr. Mair's No. 3 test, with a long-stroke cylinder.

It is inferred that, upon the whole, an admission of from 15 per cent to 20 per cent for such long-stroke cylinders is the most efficient for economy of steam.

The following, in table No. 123, are approximately the quantities of steam consumed for the most economical periods of admission, and the particular maximum pressures stated, averaged. Long-stroke engines are taken to be such as have strokes longer than two diameters. Short-stroke engines are such as have strokes less than two diameters.

Table No. 123.—Single Cylinders, not Steam-jacketed, Condensing:—Greatest Efficiency.

Cylinder.	Medium Period of Admission.	Maximum Total Pressure per square inch in the Cylinder.	Steam Consumed per Indicator Horse-power per hour.
	per cent.	lbs.	pounds.
Long-stroke....................	15 to 20	80	20
Short-stroke....................	15 to 25	73	25.2

Single Cylinders, Steam-jacketed, Condensing.—Of the Allen engine, page 504, the Sulzer engine, page 514, the Bache, 2d cylinder, page 554, the Gallatin engine, page 557, Mr. Mair's engines, page 523, and the East London engines, page 512, each using saturated steam, the particulars of performance, are here grouped together:—

Single Cylinders, Steam-jacketed, Condensing.

Engine.	Cylinder, Diameter and Stroke.	Steam-jacket.	Cut-off.	Actual Expansion-ratio.	Total Maximum Pressure in Cylinder per square inch.	Total Cut-off Pressure per square inch.	Water as Steam per Indicator Horse-power per hour.
	ins. ins.		per cent.	ratio.	lbs.	lbs.	pounds.
Allen..................	18 × 30	thorough	10	6	66	55	21.12
Do.	,,	,,	20	4.08	63	51	20.33
Do.	,,	,,	26	3.31	64	52½	19.92
Sulzer................	—	thorough	about 8	10	—	—	19.66
Bache, 2d cylinder	25 × 24	at sides	4.2	12.6	89.2	—	27.11
Do.	,,	,,	8.1	8.6	90.5	—	24.09
Do.	,,	,,	16.3	5.1	91.1	—	23.15
Do.	,,	,,	43.7	2.2	42.0	—	34.03
Gallatin..............	30.1 × 30	thorough	17.1	4.5	77.8	—	22.04
Do.	,,	,,	8.0	7.3	85.0	—	20.49
Do.	,,	,,	12.2	5.7	85.3	—	20.61
Do.	,,	,,	14.8	5.0	83.5	—	20.69
Do.	,,	,,	17.3	4.5	81.7	—	21.42
Do.	,,	,,	18.9	4.2	80.7	—	21.48
A engine, Mair.....	45 × 66	thorough	13.6	6.85	40	33	22.06
L do.	32 × 66	at sides	21.8	4.33	55.5	45.93	22.06
N do.	27 × 72	,,	30.0	3.16	55.45	44.45	21.30
O do.	27 × 72	,,	50.0	1.95	53.6	33.6	23.60
Single-acting Pumping Engines.							
E. L. W.-works:—							
Ajax	72 × 115.5	at sides	25	3.62	34.25	23.25	20.72
Cornish.........	80 × 117	,,	33	2.81	27	17.5	21.38
Wicksteed......	90 × 127	,,	25	3.66	36	23	18.82
Victoria.........	100 × 110	,,	25	3.65	40	29	20.08
K engine, Mair.....	68¼ × 96	,,	50	1.93	31	16	24.87

The lowest consumptions of steam per horse-power per hour are those effected by the thoroughly jacketed cylinders:—20 pounds, calculated for an admission of 26 per cent in the Allen engine; and 20½ pounds, as observed, in the Gallatin engine, for admissions of from 8 per cent to 14.8 per cent.

Mr. Mair's A engine uses 22.06 pounds for an admission of 13.6 per cent. Compared with the Gallatin engine, with, say, the same period of admission, the A engine uses more steam, the excess being due to the lower maximum pressure.

Of the cylinders only jacketed at the sides or case, the second cylinder of the Bache consumes, as its lowest rate, 23.15 pounds of steam per horse-power per hour, with 16 per cent admission. The most favourable result from Mr. Mair's rotative engines, is derived from the engine having the smallest diameter and the longest stroke, N of his series, cutting off at 30 per cent, and using 21.3 pounds of steam. This is also lower than the consumption of the Bache, in which the stroke is very short, although the Bache was worked with much higher maximum pressures. In the single-acting pumping engines, with comparatively low pressures, the shortest admission tabulated, 25 per cent, makes the most efficient use of the steam, 19 pounds and 20 pounds being consumed per horse-power per hour. These results are better than those of the other engines in the tablet, though the cylinders are only side-jacketed; probably because, as was before explained, there is so much wiredrawing during admission, and so high a speed of piston in the steam stroke. The calculated results of the East London pumping engines show that an admission of 20 per cent is the most economical in the consumption of steam.

It may be taken that the period of admission most favourable for efficiency, in steam-jacketed cylinders, whether thoroughly jacketed, or only side-jacketed, is from 15 per cent to 25 per cent of the stroke. The variation of efficiency for admissions between these limits appears to be small, and it is concluded that a medium admission of 20 per cent ranges amongst the most economical periods of admission. At the same time, the evidence leads to the following inferences, in which the distinction between long strokes and short strokes are the same as already defined.

1. For thoroughly steam-jacketed cylinders, having long strokes, the longer ranges of admission, within the limits above stated, are preferable; and, conversely, for those having short strokes, the shorter ranges are preferable.

2. For cylinders only steam-jacketed at the sides, the longer ranges of admission are preferable.

The following, in table No. 124, may be taken as approximately the respective maximum pressures, quantities of steam consumed, and periods of admission for greatest efficiency, leaving aside the single-acting pumping engines. The data in the first line are means of the results of the Allen and the A Mair engine; in the second line, the averages of the tests of the Gallatin, with 8.0, 12.2, and 14.8 per cent admissions; in the third line, the mean of tests L and N of Mair; and in the fourth line, the test of the Bache, 16.3 per cent admission.

Table No. 124.—Single Cylinders, Steam-jacketed, Condensing:—
Greatest Efficiency.

Cylinder.	Medium Period of Admission.	Actual Expansion-ratio.	Maximum Total Pressure per Square Inch in the Cylinder.	Steam Consumed per Indicator Horse-power per Hour.
	per cent.	ratio.	lbs.	pounds.
Thoroughly steam-jacketed, long-stroke ...	20	4	52	21
Do. do. short-stroke ...	12	6	85	20.6
Only side-jacketed, ... long-stroke	26	3.8	55.5	21.7
Do. do. .... short-stroke ...	16	5	91	23

General Conclusions on Single Cylinders.—That some adjustment is needful in order to produce directly comparable quantities, is made evident by the fact, for instance, that according to the tables, Nos. 123 and 124, long-stroke cylinders consume more steam per horse-power per hour when steam-jacketed than when not steam-jacketed, cutting off at the most effective points of the stroke. There is one obvious reason for this apparent perversion, that the maximum pressure is only 52 lbs. per square inch in the jacketed cylinder, against 80 lbs. in the unjacketed cylinder. That the efficiency is influenced by the maximum pressure is proved by the selected results from the Gallatin cylinder, in the tablet, page 558, for steam of 70 lbs. and 40 lbs. effective pressure in the boiler, showing that 2.33 pounds and 3.42 pounds more steam is used for the lower pressures than for the higher pressures. Taking the averages with steam-jacket and steamless-jacket, from table No. 118, page 557, the results correspondingly are as follows:—

Table No. 125.—Gallatin: Influence of Maximum Pressure on Efficiency.

Effective Pressure in Boiler, per Square Inch.	Nos. of Trials.	Period of Admission, Average.	Average Maximum Pressure per Sq. Inch in Cylinder.	Average Steam used per Horse-power per Hour.
lbs.	Nos.	per cent.	lbs.	pounds.
With Steam-jacket.				
40	14 to 17	14.4	55.3	23.55
70	28 to 32	14.2	83.2	20.94
Differences			27.9	2.61
With Steamless-jacket.				
40	7, 8, 9	15.8	55.0	25.57
70	24, 25, 26	15.3	78.8	23.22
Differences			23.8	2.35

Here, again, taking wide averages, the difference is marked, whether with or without steam in the jacket: about 2½ pounds more steam being used for the lower than for the higher pressure.

Again, that there is increased efficiency by steam-jacketing as against non-steam-jacketing, there is direct evidence, where the same cylinder was tried with and without steam in the jacket. A collection of cases is given in the following table:—

Table No. 126.—SINGLE CYLINDERS:—STEAM-JACKETS *versus* STEAMLESS-JACKETS.

CYLINDER.		Nos. of Trials.	Period of Admission, Average.	Average Maximum Pressure per Square Inch in Cylinder.	Average Steam used per Horse-power per Hour.
	Jacket.				
		Nos.	per cent.	lbs.	pounds.
Gallatin (short-stroke)	steam	27 to 32	14.7	82.3	21.12
	no steam	22 to 26	14.0	79.2	23.81
	Difference..........................				2.69
Bache (short-stroke)	steam	16	16.3	91.1	23.15
	no steam	13	15.5	87.7	26.25
	Difference..........................				3.10
Mair (long-stroke)	steam	2	21.8	55.5	22.06
	no steam	3	24.6	54.5	26.46
	Difference..........................				4.40

Here, the excess of steam used without the steam-jackets is 2.69 pounds and 3.10 pounds per horse-power, in the short-stroke cylinders, and 4.40 pounds in the long-stroke cylinder. For still shorter periods of admission than those given in the table, the economy by the use of the steam-jackets is still larger.

By the aid of the foregoing data, the quantities of steam used per horse-power per hour, for the same maximum pressure in the cylinder, may be estimated. Let a total maximum pressure of 80 lbs. in the cylinder be adopted as the standard. The quantities in table No. 121, page 569, for superheated steam in a non-condensing cylinder, may be accepted without modification. Those in table No. 122, page 570, for ordinary steam in unjacketed non-condensing cylinders, should be reduced to the equivalent for 80 lbs. maximum pressure. The average effective pressures above the atmosphere for 60 lbs. and 80 lbs. maximum pressures are 41 lbs. and 56 lbs., and the respective weights per cubic foot are .1759 pound and .2198 pound. The quantities of steam consumed for the two pressures are, therefore, inversely as $\frac{41}{.1759}$ and $\frac{56}{.2198}$, or directly as 255 to 233. Cutting off at 35 per cent, the steam used per horse-power is by table No. 122, 26 pounds; which is reduced, for 80 lbs. maximum pressure, to $\left(26 \times \frac{233}{255} = \right)$ 23.8 pounds. This, or, say, 24 pounds, may be adopted as approximately the quantity for direct comparison. The data for single cylinders steam-jacketed, non-condensing, page 570, are insufficient.

The quantity of superheated steam used in Hirn's condensing engine was taken, page 571, with a maximum total pressure of 65 lbs. per square inch. The equivalent quantity for a standard maximum of 80 lbs. can at best be estimated: taking the difference to be the same proportionally as was found for the Gallatin, table No. 125, page 575, where by a rise of 28 lbs. maximum pressure, the consumption of steam was reduced by 2.61 pounds, or 11 per cent off 23.55 pounds.

In raising the maximum pressure for superheated steam from 65 lbs. to 80 lbs., through 15 lbs., the percentage of reduction of weight of steam is to be diminished from 11 per cent in the ratio of 28 to 15, or to $\left(11 \times \frac{15}{28} = \right)$ 6 per cent. Then 15½ pounds of steam consumed for 65 lbs. maximum pressure, less 6 per cent, or one pound, becomes 14.5 pounds of superheated steam per horse-power per hour for 80 lbs. maximum pressure per square inch in the cylinder, cutting off at from 20 per cent to 25 per cent; the steam being superheated 150 degrees.

The data for unjacketed long-stroke condensing cylinders, table No. 123, page 573, are accepted as for the standard total maximum pressure, 80 lbs. per square inch, in the cylinder. For the short-stroke cylinders, they are to be adjusted proportionally for the standard pressure, which is 7 lbs. more than the actual pressure, 73 lbs. Guided by the difference in the Gallatin, where, by a rise of 28 lbs. maximum pressure, the consumption of steam was reduced 11 per cent, the consumption would be reduced proportionally $\left(11 \times \frac{7}{28} = \right)$ 2.75 per cent, or by $\left(25.2 \times \frac{2.75}{100} = \right)$ .7 pound. The consumption of steam is, then, $25.2 - .7 = 24.5$ pounds for the standard pressure.

For steam-jacketed condensing cylinders, the data, table No. 124, page 575, are to be similarly adjusted for the standard maximum pressure in the cylinder, 80 lbs. per square inch, resulting in a reduction of the first and third quantities, and an augmentation of the second and fourth quantities. These quantities become, in order, 18.7 pounds, 21 pounds, 19.33 pounds, and 24 pounds.

The quantities of steam consumed per indicator horse-power, reduced for a uniform maximum total pressure of 80 lbs. per square inch in the cylinder, as above estimated, are collected for comparison in the table No. 127. They are not offered as necessarily final values, but simply as the outcome of the evidence adduced.

The evidence concentrated in table No. 127 proves what has long been acknowledged,—the economical advantage of superheating the steam; and, more remarkably, the striking disadvantage of short-stroke cylinders versus long-stroke cylinders. This is a novel deduction from the evidence, and it aids in explaining apparently anomalous performances of steam engines; whilst it throws light on the still debated question of steam-jacketed cylinders versus unjacketed cylinders. The relatively large absorbing surfaces of the covers and the piston of short-stroke engines are disturbing influences which affect the operation of the steam in the cylinder to a greater extent proportionally than in long-stroke cylinders. Even if the cylinders be jacketed, the surface of the piston remains free to condense the steam during admission.

Table No. 127.—SINGLE CYLINDERS:—GREATEST EFFICIENCY.

Reduced for maximum total pressure in cylinder, 80 lbs. per square inch.

CYLINDERS.	Period of Admission.	Actual Expansion-ratios.	Steam Consumed per Indicator Horse-power per Hour.
NON-CONDENSING.	per cent.		pounds.
Superheated steam	20	4	18.5
Not steam-jacketed	35	2.5	24
Steam-jacketed	—	—	—
CONDENSING.			
Superheated steam (150 degrees)	20 to 25	4 to 3.3	14.5
Not steam-jacketed:—long-stroke	15 to 20	5 to 4	20
Do. do. short-stroke	15 to 25	5 to 3.3	24.5
Thoroughly steam-jacketed:—long-stroke	20	4	18.7
Do. do. short-stroke	12	5.6	21
Side-jacketed only:—long-stroke	26	3.24	19.33
Do. do. short-stroke	16	4.8	24

III. ECONOMY OF EXPANSIVE WORKING UNDER VARIOUS CONDITIONS:—COMPOUND CYLINDERS.

Woolf Engines, Condensing.—The leading results of compound engines on the Woolf principle, already detailed, are collected in the following table, No. 128, from Donkin's engines, pages 527 and 531; Simpson's pumping engine, page 535; Leavitt's pumping engine, page 540; and engines tested by Mr. Mair, page 565. Several of the tests noted in the table were made on engines unjacketed, or on jacketed engines in which the steam was shut off from the jackets.

Steam-jacketed Woolf Engines.—The chief contrast afforded by these engines is, as in the case of single cylinders, the variety of proportions of the stroke to the diameter of the cylinders. The variety of maximum pressures in the first cylinders is an element by which direct comparison is hindered; and there are no means afforded by the evidence itself for estimating the proper allowances to be made for differences of pressure.

The most remarkable result was obtained from Leavitt's engine, the specialities of which are a very long stroke, 5⅓ times the diameter of the first cylinder, and 2½ times that of the second cylinder—the greatest proportions in the table; the highest maximum pressure in the first cylinder, 101.3 lbs. per square inch; and the lowest consumption of steam per indicator horse-power per hour, 14.06 pounds. The cylinders are thoroughly steam-jacketed, heated by 90-lb. steam, of which the temperature is 331° F. The thorough jacketing, the high temperature of the steam in the jacket, Table No. 128.—WOOLF COMPOUND ENGINES, CONDENSING.

WOOLF ENGINES.	CYLINDERS.			Steam-jackets.	Period of Admission in 1st Cylinder.	Actual Ratio of Expansion.		Maximum Pressure per square inch in 1st Cylinder.	Steam Consumed per Indicator Horse-power.
	Diameter and Stroke.	Ratio of Diameter to Stroke.	Ratio of Cylinders.						
	ins. ins.	ratio.	ratio.		per cent.	ratios.		lbs.	pounds.
1. Donkin, factory...	13⅜ × 39⅜ 24 × 54	1 to 3 1 to 2¼	4.537	Sides and bottoms.	35	2.487 3.719	9.276	53.5	22.32
2. Do. do. ...	„	„	„	No steam in jackets.	25	3.165 3.612	11.416	49.5	32.72
3. Donkin, tandem, Hele...............	16½ × 36 30¼ × 36	1 to 2¼ to 1 1/5	3.330	2d cylinder only, side-jacketed.	13¼.........	5.089 2.590	13.185	60.5	20.55
4. Simpson, pumping engine........	28 × 66⅜ 46 × 96	1 to 2⅓ 1 to 2 1/10	3.904	Jacket inoperative.	43 top 25 bot. } 34	2.976 3.392	10.096	45.5	22.62
5. Leavitt, pumping engine............	18 × 96 38 × 96	1 to 5⅓ 1 to 2½	4.458	Thoroughly	31.5	3.025 4.096	12.391	101.3	14.06
6. Mair, C, pumping engine........	22 × 43 34 × 66	1 to 2 1 to 2	3.660	Thoroughly	21	4.280 3.071	13.133	60	17.34
7. Mair, B, pumping engine........	„	„	„	No steam............	39.2	2.524 3.071	7.750	55.5	26.62
8. Mair, D, mill engine	24½ × 41 38 × 66	1 to 1⅔ 1 to 1¾	3.870	At sides.............	36.6	2.517 3.200	8.033	79	17.73
9. Do. E, do.....	„	„	„	„	37	2.485 3.200	7.967	81	17.39
10. Do. F, do.....	„	„	„	No steam............	49.2	2.031 3.200	6.481	78	19.24
11. Mair, G, tandem..	15¾ × 51 28½ × 51	1 to 3¼ 1 to 1¾	3.270	Not jacketed........	23.3	3.526 2.715	9.571	64	21.65
12. Do. H, do.....	„	„	„	„	22.2	3.574 2.715	9.704	66	20.88

and the great length of stroke proportionally, form a combination by which the comparatively high efficiency of the engine has been attained. The evidence, already discussed, shows that the steam had been effectively dried, at least, if not superheated to some extent; seeing that, contrary to the usual order, the final weight of steam at the end of the expansion in the second cylinder, was less than the initial weight cut off in the first cylinder.

The high capacity-ratio of the second cylinder to the first cylinder of the Leavitt engine is another speciality. It is 4.458; that is, the second cylinder has more than twice the diameter of the first cylinder, with the same stroke. This high ratio has the advantage, in the present case, of extending the expansion to a high limit, in connection with a moderate degree of expansion in the first cylinder, in which the steam is only expanded three times, being cut off at 31.5 per cent.

But, so high a ratio of cylinder is not advantageous when the cylinders are not effectively jacketed. Donkin's factory engine supplies a case in point. The second cylinder is even more capacious proportionally,—being 4.537 times the first cylinder,—than in the Leavitt engine; but it only expands 3.719 times, and the steam is cut off at 35 per cent, in the first cylinder, where it expands 2.487 times. With these moderate degrees of expansion,—making together 9.276 expansions,—and a stroke three times the diameter of the first cylinder, a more efficient result would have been obtained than a consumption of 22.32 pounds of steam, but for the relatively low maximum pressure in the cylinder, with only 40 lbs. boiler pressure in the jacket, and unjacketed cylinder-covers. The temperature of the steam in the jacket did not exceed 287° F., which is 44° lower than what prevailed in the Leavitt jacket. The enormous excess-quantity of final steam in the second cylinder, page 527, or 42.76 per cent of itself, more than the sensible initial steam for expansion,—also proves how ineffective large capacity of the second cylinder may be unless the cylinder is properly jacketed. The Leavitt engine showed only a mean of 3.69 per cent final excess weight of steam.

That the insufficient jacketing of the cylinder of Donkin's factory engine,—the second cylinder especially,—led to a consumption of so much as 22.32 pounds of steam, is demonstrated by the result of not admitting steam to the jackets at all, in the second test in the table; when 32.72 pounds of steam is consumed per horse-power per hour.

The unjacketed tandem engine, Mair G and H, has a long stroke, 3¼ times the diameter of the first cylinder,—a little more than in the Donkin cylinder;—and it has a second cylinder having only 3.27 times the capacity of the first cylinder—little more than two-thirds of the capacity-ratio in the Donkin engine. Cutting off at 23.3 and 22.2 per cent in the first cylinder, in the two tests respectively, the steam is expanded there about 3½ times; in the second cylinder, 2.715 times, which is only three-fourths of the expansion in the Donkin engine, without steam in the jacket. Thus, by a moderated expansion in the second cylinder, consequent on a moderate capacity-ratio of cylinders, with 64 lbs. and 66 lbs. maximum pressure of steam, the consumption of steam is limited to 21.65 and 20.88 pounds per horse-power per hour.

With nearly equal maximum pressure, 60.5 lbs. per square inch, and a nearly equal capacity-ratio of cylinders, 3.33, Donkin's Hele engine consumes 20.55 pounds of steam, against a mean of 21.26 pounds in G and H tandem engine. As the second cylinder of the Hele engine was partially steam-jacketed, this engine should have made a considerably less consumption of steam; but for the considerably shorter stroke proportionally and the considerably shorter period of admission in the first cylinder,—13¼ per cent against 23.3 and 22.2 per cent, by which, for an unjacketed cylinder, an excessive degree of expansion was exacted.

Simpson's engine, No. 4, practically unjacketed, consumes about the same weight of steam, 22.62 pounds, as Donkin's factory engine, steam-jacketed; though the stroke is shorter proportionally, and the maximum pressure is lower, than in Donkin's engine. But, in its favour, Simpson's engine has a smaller second cylinder proportionally than Donkin's engine, and the expansion in the second cylinder is consequently carried to a less extent,—3.392 times,—than in Donkin's engine.

Turning to Mair's engines, in tests C, D, and E, the strokes do not exceed twice the diameters; but the capacities of the second cylinders are 3.66 and 3.87 times the first, allowing sufficiently good ranges of expansion there. By this facility, in connection with the higher maximum pressure in the cylinder, and thorough steam-jacketing, the consumptions of steam are limited to between 17 and 18 pounds per horse-power per hour.

From the comparisons, it appears that proportionally long strokes are conducive to economy of steam in Woolf engines, as in single cylinders; that large capacity-ratios of cylinders, that is to say, large second cylinders proportionally to first cylinders, in the ratio of from 4 to 4½ to 1, may be employed with economy, when the cylinders are thoroughly steam-jacketed; but that they are unfavourable for economy when the cylinders are only partially, or not at all, steam-jacketed, inasmuch as the extent of expansion in the second cylinder is closely related to, and regulated by, the capacity-ratio, and the larger expansions consequent upon the larger capacity-ratios are wasteful of steam; that the higher the maximum pressure of steam in a steam-jacketed cylinder, the greater is the efficiency of the steam in the cylinder, and that this greater efficiency is due, for the most part, probably, to the higher temperature maintained within the cylinder, and the better degree of dryness, or, it may be, superheat, of the steam.

When a long stroke, of at least five diameters, is combined with steam of 100 lbs. total maximum pressure in the first cylinder, and thorough steam-jacketing with steam direct from the boiler; with a second cylinder having from 3½ to 4½ times the capacity of the first cylinder, and about 12 actual expansions, the work can be done for a consumption of 14 pounds of steam per indicator horse-power per hour.

This conclusion, with others derived from the table, No. 128, are summarized in the following table, No. 129:—

Table No. 129.—WOOLF STEAM ENGINES:—EFFICIENCIES.

Stroke of 1st Cylinder in Diameters.	Total Maximum Pressure per Sq. In. in 1st Cylinder.	Steam-jacketing.	Cut-off in 1st Cylinder.	Capacity-ratio of Cylinders.	Actual Number of Useful Expansions.	Steam Consumed per Indicator H.P. per Hour.
diameters.	lbs.		per cent.	ratio.	expansions.	pounds.
5	100	thorough	30	4½	12	14
2	60	thorough	21	3½	13	17½
2	80	at sides	36	3½ to 4	8	17½
2¼	60	sides, 2d cylin. only	13	3⅓	13	20½
2	80	no steam	49	3¼	6½	19½
3	65	no steam	22	3¼	9½	21 to 21½
2½	45	no steam	34	4	10	22½

Receiver-engines, Condensing.—The chief results of tests of the receiver-engines already noticed are repeated in table No. 130, comprising Mr. Mair's J engine, page 565, with Gilain's and Weyher & Richemond's, pages 543 and 544, and the engines of the Bache, Rush, and Cher, pages 552 and 560. The J engine and Gilain's engine give the two most economical results of performance. They are both dependent upon the superheating of the steam between the cylinders for the high degree of their efficiency, in combination with their long strokes, which are more than three diameters. The capacities of the second cylinders are respectively about 3 times and 4 times those of the first cylinders, and whilst the engine J expands twice as much in the first cylinder as in the second, the Gilain engine expands but a little more in the first than in the second cylinder. This contrast results naturally from the comparatively low cylinder capacity-ratio, in engine J, against that of the Gilain engine, by which correspondingly the expansions in the second cylinder are governed, being 2.233 and 3.38 respectively.

There is no decisive evidence, in point of efficiency, in favour of either distribution of the expansion between the two cylinders—the unequal distribution in the J engine, or the nearly equal distribution in the Gilain engine. But the horse-power exerted in the J engine is nearly equally contributed by the two cylinders; whilst it is stated that in the Gilain engine the powers developed by the first and second cylinders are such as to be nearly as 3 to 1.

The Weyher & Richemond semi-portable engine supplies another instance of the value of intermediate heating of the steam, after having been exhausted from the first cylinder. Though the cylinders are jacketed only at the sides, the consumption of steam, cut off at 25 per cent in the first cylinder, and expanded 9½ times, is only 16.03 pounds per indicator horse-power per hour, which is about one pound more than the consumption of the J engine and the Gilain. The comparative shortness of the stroke of the Weyher engine may have occasioned the difference

Table No. 130.—RECEIVER COMPOUND STEAM ENGINES, CONDENSING.

RECEIVER-ENGINES.	CYLINDERS.			Steam Jacketing.	Period of Admission in 1st Cyl.	Actual Ratios of Expansion.		Maximum Pressure per sq. in. in 1st Cyl.	Steam Consumed per Indicator Horse-power.
	Diameter and Stroke.	Ratio of Diameter and Stroke.	Ratio of Cylinders.						
	ins. ins.	ratio.	ratio.		per cent.		ratio.	lbs.	pounds.
1. Simpson (Mair, J) pumping engine	21 × 66 36 × 66	1 to 3 1/7 1 to 1 5/6	2.939	Cylinders and receiver	21	4.491 2.233	10.031	75	15.19
2. Gilain, mill engine.....	27 × 89 47.2 × 118	1 to 3 1/3 1 to 2 1/2	4.050	Heated between cylinders	25	3.80 3.32	11.42	—	14.83
3. Weyher & Richemond, semi-portable	11.18 × 18.9 18.9 × 18.9	1 to 1.7 1 to 1	2.840	At sides; steam heated between cylinders	40	2.27 2.84	6.46	87	16.37
4. Do.	„	„	„	„	25	3.33 2.84	9.47	89	16.03
5. Bache, marine, No. 5	16 × 24 25 × 24	1 to 1 1/2 1 to 1	2.440	2d cylinder at sides	22.9	3.77 2.44	9.19	92.1	20.71
6. Do. No. 6	„	„	„	„	31.8	2.86 2.44	6.98	93.6	20.33
7. Do. No. 7	„	„	„	„	39.6	2.35 2.44	5.73	90.4	20.37
8. Bache, No. 1............	„	„	„	No steam	23.1	3.75 2.44	9.15	90	23.76
9. Do. No. 2...........	„	„	„	„	33.5	2.73 2.44	6.66	91	23.04
10. Do. No. 3...........	„	„	„	„	40.5	2.31 2.44	5.63	88	23.21
11. Rush, marine, No. 18	24 × 27 38 × 27	1 to 1.12 1 to .70	2.509	Thoroughly	36	2.46 2.53	6.22	82.3	18.38
12. Do. No. 19	„	„	„	„	59.8	1.60 2.53	4.03	50.2	22.10
13. Cher, marine, No. 1	43.3 × 19.7 43.3 × 19.7 (2)	1 to .45 1 to .45	2.00	Superheat 11°.5	88		2.26	41.2	23.5
14. Do. No. 2...	„	„	„	„ 16.2	60		3.21	41.7	22.1
15. Do. No. 3...	„	„	„	„ 10.8	50		3.78	41.6	20.0
16. Do. No. 4...	„	„	„	„ 18	40		4.59	42.0	22.4

Of the short-stroke engines of the Bache, Rush, and Cher, the second cylinders have only from 2 to 2½ times the capacity of the first cylinders. They comprise instances of four conditions of steam and jacketing, and the most efficient results for each case are here selected:—

Short-stroke Cylinders, Receiver, Condensing.

Condition of Steam.	Engine.	Cut-off.	Number of Expansions.	Maximum Pressure in 1st Cylinder.	Steam per Horse-power per Hour.
		per cent.		lbs.	pounds.
Thorough steam-jacketing........	Rush	36	6.22	82.3	18.38
Steam-jacketed at sides...........	Bache	31.8	6.98	93.6	20.33
		39.6	5.73	90.4	20.37
No steam in jackets...............	”	33.5	6.66	91	23.04
Superheated steam, 10°.8.........	Cher	50	3.78	41.6	20

The thorough jacketing of the Rush causes nearly 5 pounds less consumption of steam than in the Bache, without jacket-steam, although the maximum pressure is less in the Rush, and the stroke is proportionally the shorter. The thorough jacketing also makes 2 pounds less steam consumed, than the side-jacketing of the Bache. The superheating of the steam by 10.8 degrees in the Cher, also leads to a consumption less than that of the Bache side-jacketed, and 3 pounds less than the Bache with a steamless jacket, although the stroke of the Cher is less than half the diameter, and there are only 3.78 expansions.

The most effective results from receiver-engines, as from Woolf engines, are derived from long-stroke cylinders. There is no conclusive evidence of proportions of cylinders more suitable than others. A total expansion-ratio of 10 appears to be the limit of useful expansion. Second cylinders having at least 3 times the capacity of the first should be employed:—a capacity of 3½ times is probably better than 3 times that of the first cylinder, to

Table No. 131.—RECEIVER COMPOUND STEAM ENGINES:—EFFICIENCIES.

Stroke of 1st Cylinder in Diameters.	Total Maximum Pressure per Square Inch in 1st Cylinder.	Steam-jacketing.	Cut-off in 1st Cylinder.	Capacity-ratio of Cylinders.	Actual Number of Useful Expansions.	Steam Consumed per Indicator Horse-power per Hour.
diameters.	lbs.		per cent.	ratio.	expansions.	pounds.
3	80	thoroughly, receiver jacketed	21	3	10	15
1⅛	80	thoroughly	36	2½	6	18½
1¾	90	at sides; steam-jacketed between cylinders	25	3	9½	16
1½	90	at sides	32 to 40	1½	6 to 7	20½
1½	90	no steam	33	1½	6½	23
½	42	superheated 11°	50	2	4	20

admit of a more nearly equal distribution of useful expansion between the two cylinders.

The results of the J engine are the best ascertained; but they are complicated with intermediate heating of the steam. It may be taken that, with a stroke equal to three diameters of the first cylinder, for a maximum total pressure of 80 lbs. per square inch in the cylinder, and 10 actual expansions, with thorough jacketing, the work may be done with a consumption of 15 pounds of steam per indicator horse-power per hour.

This deduction, with a few others from the evidence adduced, are summarized in table No. 131, preceding page.

The engines of the Liffey and the Koning der Nederlanden, pages 546 to 549, supply instructive contrasts. In the former, the indicated expanded steam per horse-power per hour is 15 per cent greater than the steam cut off; but in the latter there is less than 3 per cent in excess; and in the 2d cylinders, there is a like contrast. The better action in the Koning is attributable to the steam-jackets on the cylinders, by which both the steam in the cylinders and the receiver steam are more effectively heated than in the Liffey; and to the compact and protective inclosure of the second slide-valve between the cylinders.

CHAPTER XI.

TESTING OF STEAM ENGINES BY THERMAL ANALYSIS.

Mr. John Farey, in 1849, gauged the rise of temperature of the condensing water, in two steam engines for the purpose of determining their relative efficiencies:—the work done, and the quantity of condensing water used being the same.[1] M. G. A. Hirn, with his late collaborator, M. O. Hallauer, reduced the process of test trials of steam engines to a system remarkable for precision and for the success with which the mechanical equivalent of heat is employed as a factor. M. Hirn had, in 1856,[2] already indicated the guiding principle of the test,—the comparison of the heat delivered to the condensing water with the heat delivered to the engine, which may be ranged as a thermo-dynamic equation. On the one side, there was the whole of the heat supplied to the engine, comprising the heat of the steam, taken as dry, passed into the cylinder, the heat conveyed by water, whether primed from the boiler, or the result of condensation in the pipes; and the heat given up by the steam condensed in the jacket. On the other side, there was the heat converted into work, the heat lost by radiation from the jacket, and the heat delivered to the condenser. M. G. Leloutre and M. O. Hallauer, under the advice of M. Hirn, jointly conducted an experimental investigation, to establish "the theory, rational and practical, of steam engines," in 1870–71; the results of which were published by them independently in 1873,[1] when for the first time the "practical theory" of M. Hirn was developed and applied in its entirety. M. Hallauer applied M. Hirn's principles to the results of many trials of Woolf engines and single-cylinder engines; and the valuations agree, within from ½ to 1 per cent of error:—affording powerful evidence of the reality as well as the utility of the mechanical theory of heat.

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[1] See a copy of Mr. Farey's "Report," in *Engineering*, for February 15, 1884, page 154.

[2] "Memoire sur la Theorie de la Surchauffe, dans les Machines à Vapeur," read October 29, 1856. *Bulletin de la Société Industrielle de Mulhouse*, 1857; page 5.

Mr. David Thomson, in 1862, recognized the utility of the temperature of the condensing water in the hot-well, as a means of testing the efficiency of condensing steam engines.[2]

Messrs. W. B. Farey and Bryan Donkin, junr., reduced to a system and published, in 1868, their method of comparing the efficiencies of steam engines, in terms of the rise of temperature of the water in the hot-well, and the quantity of the water discharged per unit of time. The quantity of heat thus rejected can be calculated; and if this quantity be divided by the number of indicator horse-power developed, the quotient is a "constant" for the engine, by which its performance can be compared with that of any other engine similarly tested. The more efficient the engine the lower is its constant. The condenser water is, for the purpose, delivered into a wooden box or trough, at one end, and it passes over and under, alternately, a series of transverse partitions, to the other end of the box, where it flows as a steady current over a notched weir or "tumbling bay." The rate of volume of water passing off is calculated in the usual manner.[3]

The observations made on the system of M. Hirn, the scope of which has been noticed above, are in three series: 1st, to determine the weight of steam consumed, and of water discharged from the condenser; 2d, to observe the temperatures; and 3d, to determine the different circumstances of the work done.

I. THERMAL ANALYSIS OF THE LOGELBACH ENGINE ON THE HIRN SYSTEM.

The case of the single-cylinder engine tested by M. Hirn, with superheated steam, which has already been dealt with (page 518), has been treated analytically by M. Hallauer.[4] The results of the eight daily trials are analysed separately, and the leading features of the trials and the analysis are here reproduced in abstract. First, the weight of steam consumed from the boiler per stroke, with the pressures in the boiler and in the cylinder at the points of cut-off and the end of the stroke, with the periods of admission, and the quantity of water injected into the condenser. There was no perceptible priming of water with the steam.

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[1] "Etude de trois Moteurs pourvus d'une Enveloppe ou Chemise de Vapeur," read April 30, 1873, by M. O. Hallauer, *Bulletin de la Société Industrielle de Mulhouse*, 1873, page 355; "Recherches Expérimentales et Analytique sur les Machines à Vapeur," presented, May 30, 1873, to the Société Industrielle du Nord de la France. See also other communications made by M. Hallauer, and published from time to time in the *Bulletin de la Société Industrielle de Mulhouse*, &c.

[2] See Mr. Thomson's observations in the discussion on his paper "On Double-cylinder Pumping Engines," in the *Proceedings of the Institution of Mechanical Engineers*, 1862; pages 285, 286. He gives the results of comparative temperatures of the hot-well, for different degrees of expansion of steam in pumping engines on a constant duty.

[3] The system of Messrs. W. B. Farey and B. Donkin, junr., is described in *Engineering* for July 17, 1868, page 58, and May 21, 1869, pages 334, 341.

[4] "Expériences Dirigées par M. G. A. Hirn, in 1873 and 1875," by M. O. Hallauer, in the *Bulletin de la Société Industrielle de Mulhouse*, 1877, page 144.

Pressure and Weights of Steam and Condensing Water.

No. of Trial.	Absolute Pressure of Steam per square inch in Boiler.	Condition of the Steam admitted to the Cylinder.	Period of Admission (approximate).	Absolute Pressure per square inch in the Cylinder.		Steam Consumed from the Boiler per stroke.	Water Injected into the Condenser per stroke.
				At Cut-off.	At End of Stroke.		
No.	lbs.			lbs.	lbs.	pound.	pounds.
1	69.55	Superheated, 448° F.	¼	60.37	13.31	.6758	20.62
2	65.96	Saturated...............	¼	53.72	13.80	.8229	20.49
3	71.00	Superheated, 419° ...	$^1/_5$	58.90	10.98	.5845	19.25
4	68.38	Superheated, 432° ...	½	32.81	11.97	.6222	18.96
5	70.66	Superheated, 383° ...	$^1/_7$	55.65	8.44	.4939	19.27
6	70.70	Saturated...............	$^1/_7$	54.53	8.16	.5808	19.65
7	71.48	Superheated, 428° ...	½	24.83	9.12	.4994	13.19
8	62.21	Superheated, 428°	¼ (without condensation)	48.83	15.72	.5984	—

Secondly, the temperatures. The temperature of the superheated steam was taken with a mercury thermometer placed in a cavity of copper or iron, closed at one end, inserted in the steam-pipe. The temperatures of the water injected into the condenser were taken with a differential air thermometer. The other temperatures, of steam in the boiler and within the cylinder, were taken, as for saturated steam, from Regnault's tables, due to the indicated pressures.

Temperatures.

No. of Test.	Temperature of Steam.					Temperature of Water in Condenser.		
	In the Boiler.	Superheated.	At the Cut-off.	At the End of Expansion.	During Exhaust.	As Discharged.	As Injected.	Difference.
No.	Fahr.	Fahr.	Fahr.	Fahr.	Fahr.	Fahr.	Fahr.	Fahr.
1	302°.27	448°	292°.93	207°.03	164°.28	88°.34	54°.68	33°.66
2	298.76	saturated	285.40	208.83	164.16	92.75	53.29	39.26
3	304	419	299.73	197.69	137.95	91.56	61.70	29.86
4	302	432	255.20	201.92	136.85	95.47	61.70	33.77
5	303.39	383	287.60	185.00	137.19	86.76	61.47	25.29
6	303.39	saturated	286.34	183.79	142.07	90.05	61.70	28.35
7	304.2	428	236.44	189.25	134.80	100.60	60.53	40.07
8	295.2	428	279.48	215.40	—	Without condensation.		

Third, the work of the engine. The net indicated work for one stroke, together with the equivalent values of this work in heat-units, are given in the following tablet. The value of the heat-unit employed in the original paper, is 425 kilogrammetres, for which the corresponding value of the English unit is 774.70 foot-pounds. This value is slightly in excess of the standard value of the English heat-unit, 772 foot-pounds; but it will be adhered to in this notice of M. Hallauer's investigations.

Work per Stroke.

No. of Test.	Net Indicated Work per Stroke.	Equivalent of Net Work per Stroke, in English Heat-units.
No.	foot-pounds.	units.
1	77,863	100.50
2	72,691	93.84
3	73,726	95.19
4	67,216	86.74
5	61,386	79.24
6	57,696	74.48
7	53,763	69.40
8	43,036	55.55

From the foregoing data, a first verification may be made of the quantity of heat brought into the cylinder, and of its equality to the sum of the net heat consumed in work, plus the heat lost by external radiation and conduction from the cylinder, plus the heat recovered in the condenser. Take, for instance, test 3, in which the steam is superheated to 419° F. The consumption of steam is at the rate of .5845 pound per stroke, taken from the boiler at the total pressure of 71 lbs. per square inch, and the temperature, according to table No. 4, of 303°.9 F. The total heat carried by one pound of the steam from the boiler is 1174.1 units, reckoned from 32° F.; and for one stroke it is

$$.5845 \text{ lb.} \times 1174.1 = 686.27 \text{ units of heat.}$$

In addition there is the superheat communicated to the steam before arriving at the cylinder, by which the temperature is raised to 418°.9 F., through (418°.9 − 303°.9 =) 115° F.; and the quantity of superheat, in calculating which M. Hallauer takes the value .50 for the specific heat of gaseous steam, is

$$.5845 \text{ lb.} \times 115^\circ \text{ F.} \times .50 = 33.61 \text{ units of heat.}$$

The total heat imported by the steam into the cylinder is, therefore, (686.27 + 33.61 =) 719.88 units per stroke, and it is now to be shown in what manner this quantity of heat is distributed.

First, the steam itself leaves the condenser as hot water at the temperature 91°.56 F., and carries off with it (.5845 lb. × (91°.56 F. − 32°) =) 34.81 units of heat, leaving (719.88 − 34.81 =) 685.07 units of disposable heat traversing the cylinder. The greater portion of this heat is received by the cold water which is injected into the condenser at the rate of 19.25 pounds per stroke. The temperature of this water is raised from 61°.70 to 91°.56, through 29°.86 F., and the quantity of heat thus consumed is (19.25 lbs. × 29°.86=) 574.76 units. Deducting this quantity from the whole disposable heat, the remainder (685.07 − 574.76=) 110.31 units has disappeared in the interval—in the form of net work done in the cylinder, and as heat lost by external cooling of the cylinder. To test the correctness of this deduction, the work done per stroke, 73,726 foot-pounds, consumes an equivalent of (73,726 ÷ 774.7=) 95.17 units of heat; and 9.92 units are, by observation, lost in cooling, by radiation and conduction. The sum of these quantities is (95.17 + 9.92=) 105.09 units, which is 5.22 units less than the 110.31 units that disappeared. That difference must be taken as error; but it is only .73 per cent, or less than 1 per cent of the total quantity of heat, 719.88 units, brought into the cylinder for one stroke.

Here is a marvellous degree of nearness to exactness in the ascertained disposal of the heat imported into the cylinder, which may be summarized as follows:—

Heat imported into the cylinder:—	Units.
1. Constituent heat of the steam as saturated, reckoned from 32° F.	686.27
2. Superheat imparted to the steam	33.61
Total heat imported	719.88

Distribution of the heat imported into the cylinder:—			Units.
Heat distributed in the cylinder	1. Effective work done in the cylinder	(13.22 per cent)	95.17
	2. External cooling	(1.38 „)	9.92
Heat rendered in the condenser	3. Steam as condensed	(4.83 „)	34.81
	4. Condensing water	(79.84 „)	574.76
	Total ascertained	(99.27 „)	714.66
	Error, by deficiency	(.73 „)	5.22

Calculated on the same principles, the respective equivalents of heat-units for the results of the other trials of M. Hirn's engine, are given in the first tablet, next page. But they are grouped differently from what has just been calculated for No. 3 trial, as explained by the headings at the side. In the first line, the gross or total quantity of heat of steam imported into the cylinder is diminished by deducting from it the residual heat in the condensed steam, or condensation water, line 2; showing the balance, line 3, being what may be called, for purposes of calculation, the net or disposable heat, which, in the case of No. 3 trial, is (719.88 − 34.81=) 685.07 units. The 4th line shows the quantity of heat absorbed by the net or effective work done in the cylinder, with the heat lost by external cooling; and the 5th line shows the difference, or excess of imported heat, in line 3, above the quantity of heat distributed in the cylinder, and by external cooling, which ought to be recovered in the condenser-water. The 6th line shows the quantity of heat actually recovered in the condensing water, exclusive, of course, of the residual heat of the condensed steam in mixture with it. These values are nearly identical with those of line 5, the errors or discrepancies being shown in the last line, 7. The errors are in all instances less than 1 per cent, except for the 7th trial, in which the error is 3 per cent—the result of some mistake in observation.

General Distribution of the Heat of Steam in the Cylinder.

No. of Trial............	1	2	3	4	5	6	7	8
	units.	units.	units.	units.	units.	units.	units.	units.
1. Total heat imported into the cylinder.................	842.45	957.70	719.88	771.21	599.76	674.56	611.47	Non-condensing.
2. Residual heat in the condensed steam in the condenser......................	38.05	49.81	34.81	39.46	27.01	33.69	27.97	
3. Heat imported into the cylinder, less the residual heat in the condensed steam in the condenser..	804.40	907.89	685.07	731.75	572.75	640.87	583.50	
4. Heat absorbed by net work done in the cylinder, and by external cooling (9.92 units)........	110.39	103.76	105.09	96.66	89.16	84.40	79.32	
5. Difference, or heat recoverable in the condensing water	694.01	804.13	579.98	635.09	483.59	556.47	504.18	
6. Heat recovered in the condensing water, exclusive of the residual heat in the condensed steam..........	693.76	804.51	574.65	640.03	487.15	557.03	521.15	
7. Error, by difference...........	+.25	−.38	+5.33	−4.94	−3.56	−.56	−16.97	

Turning now to the disturbing action of the surfaces of the cylinder and the piston, the weights of sensible steam at the point of cut-off, and at the end of the stroke, calculated in terms of the volume and pressure, are given in the following tablet for each of the eight trials, preceded by the weights of steam present in the cylinder. These are taken equal to the weights of steam consumed already given, page 520; except in the last

Action of the Cylinder-walls:—Weights of Steam in the Cylinder.

No. of Trial..........	1	2	3	4	5	6	7	8
	lb.	lb.	lb.	lb.	lb.	lb.	lb.	lb.
1. Steam directly measured present per stroke......	.6758	.8229	.5845	.6222	.4939	.5808	.4994	.6575
2. Sensible steam at cut-off	.6319	.5669	.5817	.6320	.3722	.3651	.4869	.5788
3. Difference (condensed steam)....................	.0439	.2560	.0028	−.0098 (superheated)	.1217	.2157	.0125	.0787
4. Do. per cent of steam present (line 1)..........	6.5%	30.4%	0.83%	−1.5%	24.64%	36%	2.52%	12%
5. Sensible steam at the end of the stroke........	.5949	.6156	.4820	.5401	.3883	.3764	.4203	.6575
6. Difference (condensed steam)....................	.0809	.2073	.1025	.0821	.1056	.2044	.0791	.0000
7. Do. per cent of steam present (line 1)..........	12%	25.2%	17.5%	13.2%	21.4%	35.2%	15.8%	0.0%

case, in which the steam is not condensed, but is discharged into the atmosphere, leaving a quantity of unexhausted compressed steam at the end of the return-stroke, weighing .0590 pound—nearly 10 per cent of the weight of steam consumed per stroke. In the other trials the steam was condensed, and the residue of steam and water retained after the exhaust was closed was insignificant in quantity, and is here neglected.

The differences given in the tablet between the weights of steam present and directly measured into the cylinder, and the steams in the cylinder sensible at the point of cut-off and at the end of the stroke, have already been discussed (pages 521, 522). It is remarkable that the superheated steams have so little staying power. In all but Nos. 2 and 7 tests, the steam was superheated; yet, in all of these, but No. 4, the steam was in a state of saturation at the point of cut-off, as is evidenced by the fact of the weight of sensible steam cut off, calculated as for saturated steam, being in each case less than the weight of steam consumed, except for No. 4, in which there is a slight excess—1.5 per cent—of sensible steam cut off, when calculated as saturated steam, the excess being, in fact, due to a small remainder of superheat. In this case, superheated steam was admitted for half the stroke, and it was only not entirely reduced to the state of saturation. In the similar case, of No. 7, cutting off also at half-stroke, the superheated steam was entirely reduced to the state of saturation, as is evidenced by the deficiency of sensible steam cut off. This deficiency is only 2.52 per cent; but such as it is, it results apparently from the lower temperature of superheat, 428° F., against 432° for No. 4. Taking the two cases together, it may be inferred that, when saturated steam of 70 lbs. absolute pressure per square inch, and 303° F. temperature in the boiler, was superheated to 430°, being raised 127° in temperature, the whole of the superheat was absorbed by the cylinder, and was just sufficient to protect the steam from being condensed during an admission of half-stroke.

Correspondingly, the disposal of the heat of the water and steam in the cylinder, in the progress of the stroke, may be ascertained. The temperature of the mixture is taken as the same for the water and steam. The heat that is present in the mixture consists of the heat required to raise the temperature of the whole mass as water to that of the steam, plus the heat of formation of the steam. These two heats, making together the internal heat, are present in the steam. The heat that was absorbed in external work, during the formation of the steam,—or the heat of volume,—is not, according to M. Hirn, now to be considered: it is not present as heat.

In the subjoined tablet, the first and second entries express the total internal heats in the mixture at the point of cut-off, and the end of the stroke; and the difference of these internal heats is given in the third line. In the fourth line, are the quantities of heat stored in the cylinder-walls, consisting of the heat set free by the condensation of steam during admission, plus the units of heat parted with by the total weight of steam admitted when it loses its superheat. Take, for instance, the results of the 8th test. The weight of superheated steam supplied for one stroke, is

.5984 pound, superheated to 428° F. of temperature. The temperature of the steam as cut off is 279°.48 F., and the extent of superheat is $(428 - 279.48 =)$ 148°.52 F. The quantity of superheat is, therefore, $(.5984 \times 148.52 \times .5 =)$ 44.44 units.

To this quantity is to be added the latent heat abandoned in the condensing of .0787 pound of steam during admission, calculated as follows. The absolute pressure is 48.83 lbs. per square inch, for which the latent heat is 917.4 units per pound weight. Then, $(.0787$ pound $\times 917.4 =)$ 72.22 units, the quantity of heat set free; and the sum of the superheat and the freed heat, $(44.44 + 72.22 =)$ 116.66 units, is the quantity of heat stored in the cylinder-walls during admission.

In the fifth line of the tablet are given the heat-equivalents of the absolute work of expansion, which are calculated by dividing the absolute work in foot-pounds by the mechanical equivalent of heat.

Internal Heat of the Mixed Water and Steam.

No. of Trial............	1	2	3	4	5	6	7	8
	units.	units.	units.	units.	units.	units.	units.	units.
1. Internal heat at the point of cut-off....................	701.58	690.12	636.55	674.32	437.35	453.10	528.14	649.68
2. Internal heat at the end of the stroke..................	652.50	697.54	533.17	592.90	430.76	432.79	461.40	706.18
3. Difference......................	49.08	−7.42	103.38	81.42	6.59	20.31	66.74	−56.50
4. Heat given by the steam to the surface of the cylinder	92.30	229.06	39.05	45.87	134.72	190.86	58.73	116.66
5. Heat equivalent of the absolute work of expansion	65.55	62.81	62.66	36.67	56.78	54.36	28.65	57.53

The internal heat of the mixture, it is observable, is either augmented towards the end of the stroke, or reduced, or is nearly stationary. Taking the case of the 8th trial, there are 56.50 more units of heat at the end of the stroke than at the point of cut-off; at the same time, the work of expansion has abstracted 57.53 units. In addition to these two drafts of heat, there is the heat dissipated by external cooling, taken at 2.5 French units, or 9.92 English units for one stroke. The sum of these three quantities is 123.95 units of heat consumed. The source from which this quantity of heat has been supplied is the heat stored in the metal during admission, already calculated as amounting to 116.66 units, which is only 7.29 units less than the calculated quantity demanded,—a difference which may be due to an error of observation. In any case, it is barely one per cent of the total heat imported into the cylinder; and, accordingly, the quantity of sensible steam in the cylinder at the end of the stroke, as given in the tablet, is just equal to that of the steam directly measured as present per stroke—comprising the steam shut in and compressed plus the steam admitted into the cylinder.

In cases of slight differences of initial and final heats, as in the 2d and 5th trials, the quantity of heat during expansion may be taken as nearly stationary. But during that interval, heat was converted into the work of expansion, and this heat must have been derived, not from the internal heat of the expanding steam, which was practically unchanged in quantity, but from the heat absorbed by the cylinder-walls by condensation of steam during admission. But this is only one part of the action of the cylinder-walls. Comparing, for instance, in the 5th trial, above tablet, the quantity of heat taken up by them, 134.72 units, with the outlay of heat on the work of expansion, 56.78 units; and for external cooling, 9.92 units; diminished by the small portion, 6.59 units, separated from the internal heat; it appears that of the 134.72 units, 60.11 units only are in this way appropriated, leaving 74.61 units of heat unutilized, which disappears from the cylinder at each stroke of the piston, and is recovered in the condenser.

This excess of heat received by the cylinder-walls, and thus parted with, is named by M. Hirn, *le refroidissement au condenseur*, or the cooling by the condenser,—a very important factor in the economy of the steam engine. The action is simple. When the cylinder is opened for exhaust, the steam in the cylinder rushes into the condenser, and assumes the same pressure and temperature as in the condenser. The cylinder-walls, which are hotter than the steam,—now reduced in pressure and temperature,—and which are lined with moisture, immediately part with their excess of heat, by which the superficial moisture is, in whole or in part, evaporated, and the heat thus passes into the condenser. In the second tablet, page 590, it appears from the last line, that, excepting for the last trial, there are proportions of water varying between 12 per cent and 35 per cent at the end of the stroke, caused by initial condensation. From this water the steam is generated by which the heat of the cylinder is transported to the condenser at the end of the stroke.

If the cylinder were absolutely non-conducting, so that alternate condensation and re-evaporation did not take place, the work of expansion would be exactly expressed by the difference of the internal heats at the beginning and the end of the expansion—the final heat being the less, and subtracted from the initial heat. In practice, as it has been seen, such exactness does not prevail; and in so far as the ascertained difference falls short of the work of expansion, the balance of heat for expansion-work is derived from the walls of the cylinder. The sum of the quantity of heat for expansion derived from the cylinder-walls, thus determined, plus the heat lost by external cooling, plus the heat from the cylinder-walls carried into the condenser, is equal to the whole heat absorbed by the cylinder-walls. Employing M. Hirn's symbols, these values form the equation—

$$\begin{aligned} r(M-m) &= [A F_d - (U_0 - U_1)] + 9.92 + R_c \\ &= A F_d + U_1 - U_0 + 9.92 + R_c \end{aligned}$$

in which $(M-m)$ is the weight of the condensation-water, or condensed steam, precipitated during admission; r is the latent heat of one pound of the steam; AF_d is the work of expansion in heat-units; U_0 and U_1 are the internal heats of the steam at the beginning and the end of the expansion respectively; 9.92 is the heat in units lost by external cooling; and R_c is the heat from the cylinder-walls carried into the condenser. Applying the equation to the case of the 5th trial, the values, as in the tablet, page 592, are—

$$134.72 = 56.78 - 6.59 + 9.92 + R_c;$$

and,
$$R_c = 134.72 + 6.59 - 56.78 - 9.92$$
$$= 141.31 - 66.70 = 74.61 \text{ units.}$$

The value of R_c may be verified by another line of analysis. This remnant of cylinder-surface heat is discharged together with the internal heat, U_1 of the sensible steam at the end of the stroke. It should, therefore, be found in the condenser, together with the work of back-pressure, or of expulsion of the steam from the cylinder, in addition to the internal heat, U_1. In the 5th trial, for instance, the final internal heat was found, by tablet, page 592, to be 430.76 units. Of this heat 27.01 units are retained in the condensed steam, or condensation-water, in the condenser, and the remainder, (430.76 − 27.01 =) 403.75 units, augmented by the heat sent by the work of expulsion into the condenser, 8.49 units, makes 412.24 units. Now, the heat found in the condensing water is 487.15 units, by the first tablet, page 590, and the excess, (487.15 − 412.24 =) 74.91 units, is the value of the heat R_c carried off from the cylinder-walls into the condenser. Thus:—

R_c (2d method)	74.91	units of heat.
R_c (1st method)..................................	74.61	"
Error..................	.30	

The values of the heat, R_c, carried off from the cylinder surfaces into the condenser, are given for the several trials in the following tablet; together with their proportional values as percentages of the total quantities of heat in the steam brought into the cylinder. The periods of admission and the condition of the steam admitted, are added in the last two lines.

Heat (R_c) *carried off per Stroke from the Cylinder-surfaces into the Condenser.*

No. of Trial............	1	2	3	4	5	6	7	8
	units.	units.	units.	units.	units.	units.	units.	units.
1. Values of R_c (1st method)...	65.91	148.91	69.85	80.70	74.61	146.89	86.90	−7.30
2. Do. do. (2d method)...	62.64	140.16	67.59	78.01	74.91	148.29	79.70	Without condenser.
3. Error	3.27	8.75	2.26	2.70	.30	1.39	7.20	
4. Total heat imported by the steam........................	842.45	957.70	719.88	771.21	599.76	674.56	611.47	
5. Percentage of R_c in parts of total heat................	7.8%	15.6%	9.7%	10.5%	12.43%	21.76%	14.21%	
6. Period of admission	¼	¼	1/5	½	1/7	1/7	½	¼
7. Steam superheated or saturated	sup.	sat.	sup.	sup.	sup.	sat.	sup.	sup.

The heat thus abandoned to the condenser is shown in proportions varying from 8 per cent to 22 per cent of the whole of the heat brought into the cylinder. The percentages are much greater for saturated steam than for superheated steam. Thus, cutting off at 1/7th, the percentages are, for superheated steam, 12.43 per cent; and, saturated steam, 21.76 per cent. Cutting off at ¼th, they are respectively 7.8 per cent and 15.6 per cent. It further appears that the percentage increases generally as the admission is reduced. Thus, for saturated steam, cutting off at ¼th and $^1/_7$th, the losses are 15.6 per cent and 21.76 per cent respectively: there being absolutely as much loss for $^1/_7$th as for ¼th the admission.

Since the cylinder supplies heat (R_c) at the end of the stroke, for the re-evaporation of a portion of the condensation-water, or condensed steam, the remainder only of the condensed steam is passed as water into the condenser. The quantity of this water may be calculated by comparing the heat of the steam exhausted with the heat found in the condenser. The steam exhausted consists of that which exists in the cylinder at the end of the stroke plus the steam re-formed when the exhaust is opened. Resuming the case of the 5th trial, the average back-pressure during exhaust is 2.67 lbs. per square inch, for which the total heat is 1123.77 units per pound, reckoned from the temperature 32° F. But, the temperature in the condenser is 86°.76, or 54°.76 above 32°, and the total heat reckoned from 86.76, is (1123.77 − 54.76 =) 1069.01 units per pound; or, for .4939 pound of steam, it is (1069.01 × .4939 =) 527.98 units; and this is the quantity of heat that would be delivered into the condenser, and absorbed by the condensing water, supposing that the whole of the charge of steam, .4939 pound, went there as steam; and irrespective of the heat retained by the condensed steam in the condenser, at the temperature 86°.76. But, it was seen in the first tablet, page 590, that the heat recovered in the condensing water, amounted to only 487.15 units; and the difference, (527.98 − 487.15 =) 40.83 units, indicates the absence of so much latent heat: otherwise, the discharge of a portion of the original .4939 pound of steam, as water, corresponding in quantity to 40.83 units of latent heat. The latent heat of steam of 2.67 lbs. pressure per square inch is 1018.35 units per pound, and 40.83 units is the latent heat of (40.83 ÷ 1018.35 =) .0401 pound of steam, which represents 8.11 per cent of the whole steam imported into the cylinder, passed as water into the condenser.

The weight of condensed steam delivered from the cylinder into the condenser is given in the following tablet, for each trial:—

Condensed Steam Discharged into the Condenser, per Stroke.

No. of Trial............	1	2	3	4	5	6	7	8
Weight of condensation-water from the cylinder delivered into the condenser per stroke.........	.0331	.0770	.0465	.0192	.0401	.0617	.0055	Without condenser.
Proportion per cent of the whole steam................	4.9%	9.4%	7.9%	3.11%	8.11%	10.6%	1.10%	

The leading results of the foregoing analysis of the performance of the Hirn engine, are brought together in table No. 132, supplemented by data

Table No. 132.—HIRN'S SINGLE-CYLINDER STEAM ENGINE:—LEADING RESULTS OF TRIALS MADE IN 1873 AND 1875.

1. No. of trial	1.	2.	3.	4.	5.	6.	7.	8.
2. Condition of steam; temperature	superhe'ted 448° F.	saturated	superhe'ted 419°	superhe'ted 432°	superhe'ted 383°	saturated	superhe'ted 428°	superhe'ted 428°
3. Average total pressure in boiler, per square inch	69.55 lbs.	65.96 lbs.	71.00 lbs.	68.38 lbs.	70.66 lbs.	70.70 lbs.	71.48 lbs.	62.21 lbs.
4. Period of admission (approximate)	¼	¼	1/5	½	1/7	1/7	½	¼
5. Do. do. (exact)	25.70°/。	25.70°/。	21.39°/。	45.39°/。	16.28°/。	16.28°/。	45.39°/。	28.67°/。
6. Actual ratio of expansion	3.75	3.75	4.51	2.18	5.84	5.84	2.18	3.40
7. Total pressure at cut-off in cylinder, in pounds per square inch	60.37 lbs.	53.72 lbs.	58.90 lbs.	32.81 lbs.	55.65 lbs.	54.53 lbs.	24.83 lbs.	48.83 lbs.
8. Weight of steam consumed from the boiler, per stroke	.6758 lb.	.8229 lb.	.5845 lb.	.6222 lb.	.4939 lb.	.5808 lb.	.4994 lb.	.5984 lb.
9. Total heat imported into the cylinder, per stroke units	842.45	957.70	719.88	771.21	599.76	674.56	611.47	698.09
Work per stroke:—								
10. Absolute, during admission foot-pounds	40,010	36,913	31,934	45,460	23,977	23,052	37,793	36,165
11. Do. do. heat-units	51.62	47.63	41.23	58.69	30.95	29.76	48.77	46.66
12. Do. during expansion foot-pounds	50,779	48,675	48,533	28,426	44,013	42,132	22,205	44,570
13. Do. do. heat-units	65.55	62.81	62.66	36.67	56.78	54.36	28.65	57.53
14. Do. total foot-pounds	90,789	85,588	80,467	73,886	67,990	65,184	59,998	80,735
15. Do. do. heat-units	117.17	110.44	103.89	95.36	87.73	84.12	77.42	104.19
16. Back-pressure, or work of expulsion foot-pounds	12,924	12,897	6741	6669	6604	7486	6235	37,700
17. Do. do. do. heat-units	16.67	16.62	8.70	8.62	8.49	9.64	8.02	48.64
18. Indicator, or net work foot-pounds	77,863	72,691	73,726	67,217	61,386	57,698	53,763	43,035
19. Do. do. heat-units	100.50	93.82	95.19	86.74	79.24	74.48	69.40	55.55
20. Efficiency of engine:—Indicator work, line 19, in parts of total heat imported, line 9	11.9°/。	9.80°/。	13.2°/。	11.25°/。	13.2°/。	11.0°/。	11.3°/。	7.95°/。
21. Weight of cold water injected into condenser, per stroke	20.62 lbs.	20.49 lbs.	19.25 lbs.	18.96 lbs.	19.27 lbs.	19.65 lbs.	13.19 lbs.	—
22. Initial temperature of water Fahr.	54°.68	53°.29	61°.70	61°.70	61°.47	61°.70	60°.53	—
23. Final do. do. „	88°.34	92°.75	91°.56	95°.47	86°.76	90°.05	100°.60	—
24. Heat of steam imported into cylinder, per stroke, less residual heat in condensed steam in condenser units	804.40	907.89	685.07	731.75	572.75	640.87	583.50	—
25. Heat recovered in condensing water, exclusive of residual heat in condensed steam in condenser (1st tablet, page 590) „	693.76	804.51	574.77	640.03	487.15	557.03	521.15	—

Table No. 132 (*continued*).

26. Differences of lines 24 and 25	units	110.64	103.38	110.30	91.72	85.60	83.84	62.35	—
27. Heat absorbed by net work done in cylinder, plus 9.92 units by external cooling	"	110.42	103.76	105.11	96.66	89.16	84.40	79.32	—
28. Error	"	+.22	−.38	+5.19	−4.94	−3.56	−.56	−16.97	—
Action of the cylinder-surfaces; in the distribution of the heat of the steam:—									
29. Heat given by the steam to the cylinder-surfaces (tablet, page 592)	units	92.30	229.06	39.05	45.87	134.72	190.86	58.73	116.65
30. Heat equivalent of the absolute work of expansion	"	65.55	62.81	62.66	36.67	56.78	54.36	28.65	57.53
31. Difference of internal heats at the point of cut-off and the end of the stroke	"	49.08	−7.42	103.38	81.42	6.59	20.31	66.74	−56.50
32. Absolute work, minus difference of internal heats; or difference of lines 30 & 31	"	16.47	70.23	−40.72	−44.75	50.19	34.05	−38.09	114.03
33. Heat lost by external cooling	"	9.92	9.92	9.92	9.92	9.92	9.92	9.92	9.92
34. Sum of lines 32 and 33	"	26.39	80.15	−30.80	−34.83	60.11	43.97	−28.17	123.95
35. Heat lost by the cooling action of the condenser during exhaust, or exhaust-loss; line 29 minus line 34	"	65.91	148.91	69.85	80.70	74.61	146.89	86.90	−7.30
Verification of heat lost during exhaust, line 35:—									
36. Internal heat of steam and water at the end of the stroke (tablet, page 592)	units	652.50	697.54	533.17	592.90	430.76	432.79	461.40	706.18
37. Work of expulsion of exhaust-steam into the condenser, line 17	"	16.67	16.62	8.70	8.61	8.49	9.64	8.02	48.64
38. Sum of internal heat and work of expulsion	"	669.17	714.16	541.87	601.51	439.25	442.43	469.42	754.82
39. Less the residual heat retained in the condensed steam	"	38.05	49.81	34.81	39.48	27.01	33.69	27.97	—
40. Remainder	"	631.12	664.35	507.06	562.03	412.24	408.74	441.45	—
41. Heat recovered in the condensing water, not including the heat retained in the condensed steam (line 25)	"	693.76	804.51	574.65	640.04	487.15	557.03	521.15	—
42. Difference of lines 40 and 41, the excess being the heat lost by cooling action in the cylinder during exhaust	"	62.64	140.16	67.59	78.01	74.91	148.29	79.70	—
43. Heat lost by cooling action, or exhaust-loss, as calculated otherwise, line 35	"	65.91	148.91	69.85	80.70	74.61	146.90	86.90	−7.30
44. Error, or discrepancy	"	3.27	8.75	2.26	2.69	.30	1.39	7.20	—

not specifically noted in the text. The elements of the work for one stroke of the piston are stated in lines 10 to 19, in foot-pounds and in heat-units; the indicator work, lines 18, 19, being the remainder of useful work, after deducting the back-pressure, or work of expulsion of steam remaining in the cylinder, into the condenser, lines 16, 17, from the total absolute work, lines 14, 15, consisting of the absolute work during admission and that during expansion. The efficiency of the engine, or the percentage of indicator work in parts of the total heat imported, line 9, is added in line 20. The temperatures and the disposition of the heat in the condenser are stated in lines 21 to 26. In the last of these, line 26, the differences given are the quantities by which the heats recovered in the condensing water, line 25, fall short of the corresponding sections of the heats imported into the cylinder; which should, in fact, be the heat-equivalents of the net works done in the cylinder, plus the heat lost by external cooling, the actual values of which are given in line 27. The generally near coincidence of the respective values in lines 26 and 27, is indicated by the small extent of the errors of observation in line 28. The perturbations of heat arising from the action of the cylinder-surfaces, are distinguished in lines 29 to 33. These have already been treated at pages 591 and 592. The body of heat thus played with is that which is given to the cylinder-surfaces during admission, line 29. The heat-equivalent of the absolute work of expansion, line 30, would, but for the heat-absorbing cylinder-surfaces, be measured by the difference of the internal heats at the beginning and the end of expansion. This difference does not, in fact, bear any definite relation to the heat-equivalent of expansion, being either greater or less; and the difference of these two quantities, line 32, is the resultant of the two functions, and is the measure of heat withdrawn from the store of heat in the cylinder-surfaces. The sum of this withdrawn heat and the heat lost by external cooling, line 33, is given in line 34; and it is subtracted from the heat-store in the cylinder-surfaces, line 29, showing the remainder, line 35, being the heat lost by the cooling action of the condenser during exhaust, or exhaust-loss.

The exhaust-loss is verified in the remaining lines of the table, lines 36 to 44. The sum of the internal heat at the end of expansion, line 36, and the work of expulsion, line 37, is given in line 38; and this sum diminished by the residual heat retained by the condensed steam, line 39, is stated in line 40.

The heat in the condensing water, line 41, is in excess of the remainder, line 40, by the difference, line 42, which is the exhaust-loss, and should be identical with the exhaust-loss as before reckoned, line 35.

The error, or discrepancy, line 44, is insignificant in quantity, and shows a near approach to identity.

Here ends the first exposition of the admirable system of analysis propounded by M. Hirn, and developed with a marvellous degree of success by M. Hallauer.

II. THERMAL ANALYSIS OF TWO CORLISS STEAM ENGINES.

M. Hallauer, in 1873, published comparative analyses, on the system of M. Hirn, of two Corliss steam engines of the same dimensions, one of which was unjacketed, and the other was steam-jacketed:[1] summarized as follows. They afford an instructive illustration of the utility of the system of analysis above rendered. The dimensions of the cylinders are as follows:—diameter, 20.1 inches; stroke, 39.6 inches; diameter of piston-rod, 3.15 inches; speed, 55 turns per minute. The volume described per stroke, 7.46 cubic feet; clearance-space, .253 cubic foot, being 3.39 per cent of the stroke-volume.

The Unjacketed Engine.—The steam is cut off at 4.11 per cent of the stroke, making 13.78 the actual ratio of expansion, allowing for clearance. The total pressure at cut-off is 73.32 lbs. per square inch; and at the end of the stroke, 7.13 lbs. The absolute work done for one stroke is, by the indicator, 20,726 foot-pounds. The work, calculated by means of approximate formulas, is divided as follows:—

Absolute work during admission..................	3,240	foot-pounds.
Do. do. expansion.................	17,724	„
Total...............	20,964	„

The back-pressure is 1.77 lbs. per square inch, equivalent in work to 1903.5 foot-pounds; and deducting this value, the net indicated work is (20,726 − 1903.5 =) 18,822.5 foot-pounds per stroke. The back-pressure of compression is 1.46 lbs. per square inch; and as only .0026 pound of exhaust-steam is shut in, it is neglected in the calculations for heat. The weight of steam and water passed from the boiler into the engine, is .2474 pound per stroke, of which .011 pound, or 4.4 per cent, is water. The remainder, .2364 pound of steam, is partly condensed during admission:—

	Pound.		Per Cent.
Steam at the point of cut-off................................	.0933	or	37.7
Steam condensed during admission, being water at the point of cut-off....................................	.1431	or	57.9
Steam admitted..	.2364	or	95.6
Priming..	.0110	or	4.4
Steam and water admitted into the cylinder...............	.2474	or	100.0

At the end of the stroke, there is .1449 pound of steam, and .1025 pound of water, respectively 58.6 per cent and 41.4 per cent:—together .2474 pound. The steam consumed per absolute indicator horse-power per hour is 23.62 pounds.

The total internal heat at the point of cut-off, is the sum of the water-heat of the mass of steam and water, and the heat of formation of the steam. It amounts to 144.83 units. At the end of the stroke, the total internal heat amounts to 169.40 units. The difference is 24.57 units. The absolute work of expansion is 22.54 units, and the loss of heat by external cooling is 4.96 units.

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[1] *Bulletin de la Société Industrielle de Mulhouse*, 1873, page 592.

Difference of initial and final internal heats	24.57	units.
Absolute work of expansion	22.54	„
External cooling	4.96	„
	52.07	„

The heat lodged in the cylinder-surfaces during admission, by condensation of steam, is the latent heat of this condensed steam. It is 128.53 units; making, with 1.59 units for friction of the piston, a total of 130.12 units; and (130.12 − 52.07 =) 78.05 units is the heat carried by evaporation into the condenser.

The Steam-jacketed Engine.—The steam is cut off at 7.78 per cent of the stroke, making the actual ratio of expansion 9.11. The total pressure at the end of the cut-off is 74.30 lbs. per square inch; and at the end of the stroke it is 11.41 lbs. per square inch. The absolute work done for one stroke is 30,353 foot-pounds, distributed, as in the previous case, as follows:—

Absolute work during admission	6,355.5	foot-pounds.
Do. do. expansion	23,728	„
Total	30,083.5	„

The back-pressure, 3.31 lbs. per square inch, gives rise to the negative work of resistance, 3555 foot-pounds, which reduces the absolute work to the net indicator work, (30,353 − 3555 =) 26,798 foot-pounds for one stroke.

The weight of steam and water consumed from the boiler amounts to .2763 pound per stroke; of which .0106 pound, or 3.83 per cent, is condensed in the jacket. The rest, .2657 pound, passes into the cylinder, as .2518 pound of steam and .0139 pound of priming water, which is 5 per cent of the mixture. The steam compressed is .0044 pound in weight. The weight of steam present at the cut-off, not including the .0044 pound of steam compressed, is .1422 pound; the rest of the steam introduced, (.2518 − .1422 =) .1096 pound, is reduced to the state of water:—

	Pound.		Per Cent.
Steam at the point of cut-off, per stroke	.1422	or	53.5
Steam condensed during admission	.1096	or	41.3
	.2518	or	94.8
Priming	.0139	or	5.2
Steam and water in the cylinder	.2657	or	100.0
Steam condensed in the jacket	.0106	or	3.99
Total consumed from the boiler	.2763		

At the end of the stroke, there is .1021 pound of steam, and .0184 pound of water; respectively 85 per cent and 15 per cent of the total quantity, .2657 pound, passed through the cylinder. The steam consumed per absolute indicator horse-power is 18.02 pounds per horse-power per hour.

The total internal heat amounts to 190.15 units at the point of cut-off, and to 248 units at the end of the expansion.

	Units.
Difference of initial and final internal heats	57.85
Absolute work of expansion	30.95
External cooling	5.55
	94.35

The heat lodged in the cylinder-surfaces during admission is the latent heat disengaged by the condensation of .1096 pound of steam, as before stated, amounting to 98.36 units. Adding 9.48 units by external cooling, and 1.59 units for the friction of the piston, the sum of these, 109.43 units, is the store of heat, from which 94.35 units is provided. The remainder, (109.43 − 94.35 =) 15.08 units, is the heat lost by evaporation into the condenser.

Interpretation.—First of all, it is shown that, in the unjacketed cylinder, continuous evaporation takes place from the cylinder-surfaces during expansion; since there is 57.9 per cent of condensation-water at the end of the admission, and only (41.4 − 4.4 (priming) =) 37.0 per cent at the end of the stroke; showing that (57.9 − 37.0 =) 20.9 per cent of water has passed into the state of steam. In the jacketed cylinder, the proportion of steam condensed during the admission is much less. It is 41.3 per cent against 57.9 per cent; and at the end of the stroke there is only (15 − 5.2 (priming) =) 9.8 per cent. The proportion evaporated during expansion is (41.3 − 9.8 =) 31.5 per cent against 20.9 per cent in the unjacketed cylinder. The augmentation of evaporation during the stroke, here shown, brings with it useful work done on the piston, and the advantage of a much smaller proportion of water in the cylinder at the end of the stroke:—15 per cent against 41.4 per cent, including priming; for there is thus a much smaller withdrawal of heat from the cylinder-surfaces by evaporation into the condenser. In fact, only 15.08 units of heat were thus lost from the jacketed cylinder; whilst 78.05 units were lost from the unjacketed cylinder. The immediate consequence of such cooling of the cylinder at the end of the stroke is that the heat thus lost is to be reinstated during admission for the succeeding steam-stroke.

The broad result of the comparative performances of the two engines is, that whilst the unjacketed cylinder consumes 23.63 pounds of steam per absolute horse-power per hour, comprising, say, 4.5 per cent of water, the jacketed cylinder consumes only 18.02 pounds, with, say, 5 per cent of water:—being 5.61 pounds, or 23.75 per cent less steam. The economy here attained is due solely to the action of the jacket, according to M. Hallauer. But this can scarcely be the case, seeing that the expansion was carried much further—uneconomically so—in the unjacketed than in the jacketed cylinder; and that part of the economy is no doubt due to the more suitable period of admission in the jacketed cylinder.

The relative economy is demonstrable also in terms of the heat-equivalents. In the unjacketed cylinder, the constituent heat of the steam, which is 95.5 per cent of the mixture, plus the heat in the 4½ per cent of water, amount together to 26,449.51 units per absolute horse-power per hour; and the combined heat for the jacketed cylinder amounts to 20,094.50 units, making a reduction of 6355.01 units, or 24.03 per cent.

Comparing now the disposition of heat per stroke of the piston, in the unjacketed cylinder, .2474 pound of steam, of which 4.5 per cent is water, is consumed, carrying 280.82 heat-units into the cylinder. The total internal heat at the end of the stroke was found to be 169.40 units, making a difference of 111.42 units, which would have been entirely utilized in work, if there had not been either loss or cooling externally or by the condensers. But, in fact, the net indicated or useful work was 18,822.5 foot-pounds, or 24.30 heat-units,[1] which is only 21.8 per cent of the quantity of heat that disappeared.

In the jacketed cylinder, similarly, .2657 pound of steam, comprising 5 per cent of water, passes through the cylinder for one stroke, carrying 312.25 units of heat into the cylinder. The total internal heat at the end of the stroke is 248 units, and the difference is 64.25 units, of which 26,798 foot-pounds, or 34.59 units, was utilized, being 53.8 per cent of the heat that disappears. This is in advance of the proportional utilization in the unjacketed cylinder, to the extent of (53.8−21.8=) 32.0 per cent gain.

The respective efficiencies of the two cylinders, in terms of the total heat passed into the cylinder per stroke, and the heat utilized, are as follows:—

$$\text{Unjacketed cylinder, } \frac{24.30 \times 100}{280.82} = 8.65 \text{ per cent efficiency.}$$

$$\text{Jacketed cylinder, } \frac{34.59 \times 100}{312.25} = 11.08 \quad \text{do.} \quad \text{do.}$$

III. ANALYSIS OF WEYHER AND RICHEMOND'S COMPOUND ENGINE.

M. Hallauer[2] analyses the work of Weyher & Richemond's compound engine, at Colmar, already noticed, page 544. The analysis is here given, in abbreviated form, of the performance during the first trial, as an example of M. Hallauer's method of treating compound engines. A series of indicator diagrams, thought to be the best, was selected for the purpose of calculation.

Work for One Stroke.

1st Cylinder:—	Foot-pounds.
Absolute work during admission	5,707
Do. do. expansion	5,678
Do. do. total	11,385
Work of back-pressure	6,821
Net indicated work	4,564

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[1] Using the divisor 774.70, the equivalent of the French divisor, 425 kilogrammes.

[2] *Bulletin de la Société Industrielle de Mulhouse*, 1881; page 156.

2d Cylinder:—	
Absolute work of expansion	11,400
Work of back-pressure	1,533
Net indicated work	9,866
Total indicated work of the two cylinders	14,430
Total absolute work of the two cylinders (absolute work in 1st cylinder, plus absolute work in 2d cylinder, minus work of back-pressure in 1st cylinder)	15,963
Turns per minute	88.5
Indicator horse-power	77.42
Ratio of expansion	7.58

The average power during the trial was 78.11 horse-power, which is a very little more than the power deduced from the selected diagrams. The steam consumed from the boiler per horse-power per hour was 16.61 pounds, comprising 3.7 per cent of priming water: equivalent, for 77.42 horse-power, and 88.5 turns per minute, to .1228 pound per stroke, consisting of .1183 pound of steam, and .0045 pound of water. The total heat carried into the cylinder by this mixture is 141.26 units. To check this deduction, the heat gained in the condenser is to be ascertained. Cold water at 48° F. was injected at the rate of 2.321 pounds per stroke, and was heated to 96°.73, rising in temperature 48°.73. The heat thus acquired is $(2.321 \times 48.73 =)$ 113.10 units. The condensed steam in the condenser has the same temperature, 96°.73, or 64°.73 above 32°, and retains $(.1228 \times 64.73 =)$ 7.95 units of heat. The heat lost by the steam was, therefore, $(141.26 - 7.95 =)$ 133.31 units, of which the injection-water only absorbed 113.10 units, leaving $(133.31 - 113.10 =)$ 20.21 units consumed by the indicated work plus the external cooling. The indicated work, 14,430 foot-pounds per stroke, is equivalent to (14,430÷774.70[1]=) 18.63 units of heat, and adding the estimated loss by cooling, 2 units, the sum, 20.63 units of heat, is only 0.42 unit more than 20.21 units, already calculated for the purpose. The error, .42 degree, is but 0.3 per cent of the heat brought into the cylinder.

The indicated work, 18.63 heat-units per stroke, is 13.19 per cent of the whole heat imported from the boiler.

The trial at the brake gave 67.56 horse-power, which is 87.5 per cent of the indicator horse-power.

Proceeding to the consideration of the perturbations of heat in the engine, there is, in the first place, the charge, .1228 pound of a mixture of steam and water, measured from the boiler, already mentioned, plus .0163 pound of compressed steam: together, .1391 pound, which comprises the water formed in the steam-jacket, as, by the formation of the jacket, before explained, water is not deposited in it, but is carried into the first cylinder. The charge, .1228 pound, consists of .1183 pound of steam and .0044 pound of water, and, with the compressed steam, there is $(.1183 + .0163 =)$ .1346 pound of steam and .0044 pound of water; together, .1391 pound.

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[1] The equivalent of the value employed by M. Hallauer.

At the end of the admission, the .1391 pound of steam and water is composed of .0979 pound of steam deduced in terms of the volume, .4732 cubic foot, and the absolute pressure, 89.65 pounds per square inch, and .0412 pound of water, which is 29.7 per cent of the mixture. This water, less the priming, or (.0412−.0044=) .0368 pound, is the result of condensation during admission; and this condensation has set free 32.93 heat-units, which are lodged in the cylinder-surfaces, to be partly restored by re-evaporation during expansion. At the end of the stroke, accordingly, the .1391 pound of mixture contains more steam and less water than at the end of the admission:—namely, .1180 pound of steam, calculated from the volume, 1.130 cubic feet, and the pressure, 43.38 lbs. per square inch; and .0211 pound of water, being 15.2 per cent, against 29.7 per cent at the beginning of the expansion. Of the steam present at the end of the stroke, a fraction is detained and compressed, namely .0163 pound, as before stated, leaving (.1391−.0163=) .1228 pound of mixed steam and water to pass to the second cylinder:—this being the weight of the charge from the boiler per stroke.

The expansion in the intermediate space and the second cylinder leads to but little change in the elements. There is present in the second cylinder the charge from the boiler, .1228 pound, plus .0073 pound of steam compressed in the clearance:—making together, .1301 pound; and this mixture, at the end of the stroke, consists of .1098 pound of steam, as deduced from the final volume, 3.203 cubic feet, and the final pressure, 13.23 lbs. per square inch; and .0203 pound of water, 15.6 per cent.

The internal heats of the mixture of steam and water, are,—

At the end of admission in the first cylinder.........	119.43 heat-units.
At the end of the stroke in the first cylinder.........	133.32 „
At the end of the stroke in the second cylinder.....	121.03 „

From these are to be deducted the internal heats of the compressed steam, which, for the first cylinder, is 13.89 units; and, for the second cylinder, 6.75 units. The difference of the internal heats at the end of admission, and the end of the stroke of the second cylinder, thus corrected, is,—

$$(119.43 - 13.89) - (121.03 - 6.75) = -8.74 \text{ units.}$$

The final internal heat is greater than the initial.

Arrived at the end of the stroke, the steam is exhausted from the second cylinder into the condenser; and, at the same time, a part of the water deposited on the cylinder-surfaces is evaporated.

The absolute work of expansion demands.............	16.67 units.
The external cooling uses................................	1.98 „
The internal heat is augmented........................	8.74 „
Total..................	27.39 „

This heat could only be supplied by the condensations in the jacket and in the first cylinder during admission, which yielded, in all, 32.93 heat-units.

The excess of this over the heat taken up, is (32.93 − 27.39 =) 5.54 units, or 3.9 per cent of all the heat supplied from the boiler; and this is the lost heat, carried off by evaporation into the condenser.

To deduce the lost heat independently from the heat in the condensing water:—The final internal heat, minus the heat that remains in the cylinder with the compressed steam, plus the heat sent to the condenser by the work of expulsion of exhaust-steam, or the exhaust back-pressure, represents the total heat that would be found in the condensing water, if there were no lost heat by final evaporation.

Final internal heat	121.03	units.
Heat in compressed steam, deduct	6.75	„
Net internal heat	114.28	„
Work of expulsion, add	1.98	„
Total heat	116.26	„

Now, the injection-water gains 113.10 units of heat, whilst the condensed steam retains 7.95 units; together, 121.05 units; showing (121.05 − 116.26 =) 4.79 units, the heat lost by evaporation into the condenser. The two values of the lost heat are,—

By the first calculation	5.54	units.
By the second calculation	4.79	„
Difference	.75	„

This difference, by error, is little more than ½ per cent of the heat, 141.26 units, imported from the boiler.

In the foregoing analyses, on M. Hirn's system, of the performance of steam engines, the first analysis, by M. Hallauer, showing the distribution of the heat of the steam, in the Hirn engine, in general terms, is very instructive. The following are the terms arranged in an equation:—

Total heat imported into the engine, per stroke,
= net work done per stroke, in heat-units
+ heat lost by external cooling
+ residual heat in condensed steam
+ heat recovered in the condensing water itself.

More generally, the equation may stand thus:—

Total heat imported into the engine, per stroke,
= net work
+ external cooling
+ total heat recovered in the condenser.

But, in these equations, the distribution of the heat recovered in the condenser is not traced. Overlooking, meantime, the minute perturbations of heat caused by compression, and classing the work of compression with the back-pressure of exhaust, the total heat recovered in the condenser, is equal to the internal heat of the steam and water in the cylinder at the end of the stroke, plus the heat equivalent to the back-pressure of exhaust or expulsion of steam from the cylinder, plus the heat lost by evaporation into the condenser; and the equation becomes,—

Total heat imported into the engine, per stroke,

$$\begin{array}{l}\left.\begin{array}{l}= \text{net work}\\ + \text{ exhaust back-pressure}\end{array}\right\}\text{total absolute work}\\ + \text{ external cooling}\\ + \text{ internal heat of steam and water at the end of the stroke}\\ + \text{ loss by evaporation into the condenser.}\end{array}$$

Mr. Michael Longridge published, in 1881, a detailed thermal analysis of the performance of Mr. E. Heyworth's engine and boiler, at Blackburn.[1] This appears to have been the first published example of the employment in England of M. Hirn's method of analysis.

IV. SINGLE AND COMPOUND STEAM ENGINES, TESTED BY MR. J. G. MAIR.

The results of analyses for a single-cylinder condensing engine, A in the original paper, and for the compound receiver-engine J, already noticed, page 583, are selected for illustration, from Mr. Mair's papers already referred to;[2] and are here given in detail, with the symbols employed by Mr. Mair:—

Table No. 133.

THERMAL ANALYSIS, ENGINES A AND J.—MR. J. G. MAIR. 1881.

DATA.	A	J
Date of trial	Feb. 1881	Oct. 1881
Type of engine	single-cylinder beam	co'pound receiver beam
Time running	6 h. 46 m.	6 h. 0 m.
Total number of revolutions, from counters	5938	8632
Revolutions per minute	14.62	23.98
Indicator horse-power I.H.P.	101.79	127.40
Total horse-power H.P.	115.04	138.30
Feed to boilers, deducting steam-pipe drains lbs.	15,840	12,032
Temperature of feed-water	66° F.	59°.05 F.
Temperature of injection-water i	—	50°
Temperature of air-pump discharge f	70°	73°.4
Air-pump discharge per minute lbs.	—	1225.8
Water drained from jackets lbs.	648	1605
Percentage of priming water in steam per cent	4	4
Height of barometer during trial inches	29.60	30.20

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[1] *Trial of the Engines and Boiler at E. Heyworth's, Esq., Audley Hall Weaving Shed, Blackburn.* 1881.

[2] See foot-note, page 522.

Table No. 133 (*continued*).

Data.	A	J
Volumes in Cubic Feet, and Weights per Stroke, in Pounds.		
Volume of 1st cylinder V_2	60.346	13.02
Volume of clearance of 1st cylinder Vo	1.000	.2151
Volume of 2d cylinder V	—	38.66
Volume of clearance of 2d cylinder Vo_1	—	1.035
Volume at end of admission V_1	8.953	2.916
Total number of expansions	6.85	13.61
Weight of steam, M, and water, M_P, through cylinder	1.2798	.60768
Weight of priming through cylinder M_P	.05335	.02430
Weight of steam condensed in jackets M_J	.05400	.08925
Weight of steam and water shut in 1st cyl. clearance mc	.01440	.05913
Total weight of steam and water in 1st cyl., $M + M_P + mc$	1.29420	.66681
Weight of steam at end of admission m_1	.72700	.43920
Do. do. 1st cylinder stroke, m_2	.92632	.51789
Weight of steam and water shut in clearance of 2d cylinder mc_1	—	.01680
Total weight of steam and water in 2d cyl., $M + M_P + mc_1$	—	.62448
Weight of steam at end of 2d cylinder stroke m_3	—	.50015
Weight of air-pump discharge $\Pi + M + M_P$*	—	25.558
Work Expressed in Thermal Units, per Stroke.		
Absolute work of full admission Wa	55.16	37.49
Absolute work of total expansion in 1st cylinder Wb	117.00	54.61
Work of expulsion from 1st cylinder We	21.56	31.47
Work of compression in 1st cylinder Wd	1.80	7.14
Absolute work in 2d cylinder Wb_1	—	72.69
Work of expulsion from 2d cylinder We_1	—	12.03
Work of compression in 2d cylinder Wd_1	—	0.60
Total absolute work of expansion $Wb + Wb_1 - We$	117.00	95.83
Indicated work in 1st cylinder Wi H.P.	148.80	53.49
Indicated work in 2d cylinder Wi L.P.	—	60.06
Total indicated work Wi	148.80	113.55
Radiation R	15.00	22.72
Heat Passing Through the Engine per Stroke.		
Thermal units from boiler, as dry steam $M \times H$†	1431.0	686.06
Do. do. as priming water $M_P \times h$‡	13.2	6.76
Do. do. by condensation in jackets $M_J \times L$§	49.6	80.09
Total heat through engine per stroke Q	1493.8	772.91

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* Π = Weight of injection-water.

† M = steam through cylinders per stroke. H = total heat of steam, measured from 32° F.

‡ M_P = priming water through cylinder per stroke. h = thermal units necessary to raise 1 lb. of water from 32° F. to the temperature t.

§ M_J = effective condensation in jackets per stroke. L = latent heat of evaporation.

Table No. 133 (*continued*).

DATA.	A	J
VERIFICATION OF Q.		
Heat retained in condensed steam $(M + M_p)(f - 32)$...Q_1	—	25.16
Heat absorbed by injection-water Π* $(f - i)$............Q_2	—	583.83
Indicated work in thermal units..........................W_i	—	113.55
Radiation ....R	—	22.72
Error	—	27.65
	—	772.91
Percentage of error in parts of total heat Q................	—	3.57
INTERNAL HEAT OF STEAM AND WATER CONTAINED IN CYLINDERS.		
At point of cut-off...I_1	915.60	540.80
At end of stroke of 1st cylinder...........................I_2	1035.30	583.70
Do. do. 2d do. ....I_3	—	546.82
At beginning of compression in high-pressure cylinder..Ic_1	12.88	63.54
At end of do. do. do. ..Ic_2	15.32	33.36
At beginning of compression in 2d cylinder............Ic_3	—	10.60
At end of do. do. ....Ic_4	—	17.89
HEAT GIVEN UP TO CYLINDER-WALLS DURING ADMISSION.		
Heat through engine per stroke ....Q	1493.80	772.91
Heat given up in jackets...............................M_jL	49.60	80.09
	1440.20	692.82
Internal heat of steam and water shut in 1st cyl. clearance, Ic_2	15.32	33.36
	1459.52	726.18
Absolute work of admission ....Wa	55.16	37.49
Internal heat of steam and water at end of admission...I_1	915.60	540.80
Heat given up on admission ....Qa	488.76	147.89
	1459.52	726.18
HEAT GIVEN UP TO CYLINDER-WALLS DURING COMPRESSION.		
Internal heat of steam and water shut in 1st cylinder by compression ....Ic_1	12.88	63.54
Work of compression in 1st cylinder, in thermal units..Wd	1.80	7.14
	14.68	70.68
Internal heat at end of compression in 1st cylinder....Ic_2	15.32	33.36
Heat given up to 1st cylinder-walls during compression..Qd	− 0.64	37.32
Internal heat shut in 2d cylinder by compression ......Ic_3	—	10.60
Work of compression in 2d cylinder, in thermal units, Wd_1	—	0.60
	—	11.20
Internal heat at end of compression in 2d cylinder....Ic_4	—	17.89
Heat given up to 2d cyl.-walls during compression...Qd_1	—	− 6.69

*Π = Weight of injection-water.

Table No. 133 (*continued*).

DATA.	A	J
HEAT ABSORBED BY STEAM DURING EXPANSION.		
Difference in internal heats in 1st cylinder..........$I_2 - I_1$	119.7	42.90
Absolute work of total expansion in 1st cylinder......Wb	117.0	54.61
Heat absorbed during expansion in 1st cylinder.......Qb	236.7	97.51
Internal heat at end of 2d-cylinder stroke ....I_3	—	546.82
Internal heat shut in 1st cylinder by compression ......Ic_1	—	63.54
Work of total expansion in second cylinder...........Wb_1	—	72.69
	—	683.05
Internal heat of steam shut in 2d-cylinder clearance...Ic_4	—	17.89
	—	665.16
Internal heat at end of 1st-cylinder stroke + work of expulsion from 1st cylinder.......................$I_2 + We$	—	615.17
Heat absorbed during 2d-cylinder stroke ....Qb_1	—	49.99
EXHAUST WASTE (OR LOSS) FROM 1ST CYLINDER. BY DIRECT MEASUREMENT.		
Internal heat at end of 1st-cylinder stroke................I_2	1,035.30	583.70
Internal heat of steam shut in 1st-cylinder by compression..Ic_1	12.88	63.54
	1,022.42	520.16
Work of expulsion from 1st cylinder, in thermal units..We	21.56	31.47
Indicated work in 1st cylinder	148.80	53.49
Radiation from 1st cylinder................................R_H	15.00	7.00
Heat given up by condensation in 2d cylinder and receiver jackets............................$M_JL_1 + M_JL_R$*	—	47.66
Exhaust waste (or loss) from 1st cylinder into 2d cyl...Qc	286.02	113.13
Total heat through engine..................................Q	1493.80	772.91
OTHERWISE.		
Heat given up to surfaces of 1st cylinder during admission...Qa	488.76	147.89
Heat given up during compression ....Qd	−0.64	37.32
Heat given up by condensation in 1st-cyl. jacket...M_JL_R	49.60	32.43
Total heat given up to surfaces of 1st cylinder............	537.72	217.64
Heat absorbed by steam during expansion in 1st cylinder + radiation....................................$Qb + R_H$	251.70	104.51
Exhaust waste (or loss) from 1st cylinder into 2d cylinder..Qc	286.02	113.13
Total heat given up from walls of 1st cylinder............	537.72	217.64

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* M_JL_1 = condensation in 2d-cylinder jacket. M_JL_R = condensation in receiver-jacket.

Table No. 133 (*continued*).

DATA.	A	J
EXHAUST WASTE (OR LOSS) FROM 2D CYLINDER.		
Internal heat at end of 2d-cylinder stroke................I_3	—	546.82
Internal heat of steam shut in low-pressure cylinder by compression..Ic_3	—	10.60
	—	536.22
Work of expulsion from 2d cylinder, in thermal units..We_1	—	12.03
Indicator horse-power + radiation, in ther. units...$Wi + R$	—	136.27
Exhaust waste (or loss) by evaporation into condenser..Rc	—	88.39
Total heat through engine per stroke......................Q	—	772.91
OTHERWISE.		
Heat given up from walls of 1st cylinder during supply to 2d cylinder...Qc	—	116.83
Heat given up by compression..........................Qd_1	—	− 6.69
Heat given up by condensation in 2d-cyl. jacket...M_1L_1	—	37.24
Total heat given up to walls of 2d cylinder................	—	147.38
Heat absorbed by steam during 2d-cylinder stroke + radiation...$Qb_1 + R_1$	—	58.99
Loss by cooling action of condenser (evaporation into condenser)...Rc	—	88.39
Total heat given up from walls of 2d cylinder............	—	147.38
CONSUMPTION OF COAL PER INDICATOR HORSE-POWER PER HOUR.		
Total heat through engine per stroke ($Q - Q_1$),* calculated from the temperature of the air-pump discharge	1445.20	747.75
Thermal units per indicator horse-power per minute, calculated from the temperature of air-pump discharge...	415.10	281.50
Pounds of coal per indicator horse-power per hour, supposing feed-water, taken from hot-well, and coal to give up 11,000 thermal units per pound............	2.26 lbs.	1.53 lbs.
DISPOSAL OF HEAT ($Q - Q_1$) PASSING THROUGH ENGINE PER STROKE.		
Indicated work, per cent......................................	10.3	15.2
Rejected heat and error, per cent............................	88.7	81.8
Radiation, per cent...	1.0	3.0
SUMMARY OF RESULTS.		
Absolute boiler pressure in lbs. per square inch............	48	76
Revolutions per minute...	14.62	23.98
Speed of piston per minute, in feet..........................	160.82	263.78
Total number of expansions	6.85	13.61
Type of engine..	Single-cyl.	Receiver

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* Q_1 = the residual heat in the condensed steam in the condenser.

Table No. 133 (*continued*).

Data.	A	J
Summary of Results (*continued*).		
Percentage of supply from boiler passed through jacket...	4.0	13.2
Indicator horse-power	101.79	127.40
Total horse-power..	115.04	138.30
Percentage of loss by back-pressure.........................	11.5	7.9
Percentage of water present in cylinder at cut-off.........	43.8	34.1
Do. do. at end of 1st-cylinder stroke..........	28.4	22.3
Do. do. at end of 2d-cylinder stroke..........	—	19.9
Percentage of total heat through engine lost by R*c* (or evaporation into the condenser).........................	19.1	11.4
Thermal units per indicator horse-power per minute......	429.11	291.0
Thermal units per total do. do.	379.8	268.0
Pounds of dry saturated steam per indicator horse-power per hour	22.06	14.84
Pounds of dry saturated steam per total horse-power per hour..	19.52	13.67
Maximum theoretical efficiency*.............................	.281	.305
Actual efficiency†...	.099	.147
Thermal units absorbed by injection-water per indicator horse-power per minute	—	219.8
Donkin's coefficient	—	225.1

* $\frac{T_1 - T_2}{T_1}$: T_1, absolute temperature in boiler; T_2, that of condenser water from air-pump.

† Actual efficiency = $\frac{33000}{772}$ ÷ thermal units per indicator horse-power.

CHAPTER XII.—FRICTIONAL RESISTANCE OF STEAM ENGINES.

The frictional resistance of steam engines, in general, varies from 20 per cent to 8 per cent or 10 per cent.

The results of trials of portable engines at Cardiff, in 1872, already noticed, page 498, in which the horse-power delivered at the brake applied on the driving shaft, was measured and compared with the indicator power, showed the following results—in the order of efficiency: that is, the ratio of the work transmitted through the engine to the fly-wheel shaft and available for doing external work, to the effective work done by the steam in the cylinder. Only such results are here brought together in table No. 134, as were obtained under normal conditions of engine, and of distribution of steam.

Great variation is found in the frictional resistances:—from 12 per cent to 20 per cent, giving from 88 per cent to 80 per cent of the indicator power at the fly-wheel shaft. The indicator powers range from 13 to 20 horse-power; and with the three higher powers the lower percentages of power are delivered at the fly-wheel shaft. The speeds likewise vary very much:—

from 114 to 180 turns per minute, the higher speeds showing the lower efficiencies of transmission.

Table No. 134.—FRICTIONAL RESISTANCE OF PORTABLE ENGINES OF 8 NOMINAL HORSE-POWER. Cardiff, 1872.

Order No.	Cylinder: Diameter and Stroke.	Effective Mean Pressure in Cylinder per Square Inch.	Turns per Minute.	Indicator Horse-power.	Frictional Resistance.		Efficiency of Transmission at Brake.	
	ins. ins.	lbs.	turns.	I.H.P.	H.P.	per cent.	H.P.	per cent.
1	9 × 12	29.72	123.7	14.0	1.7	12	12.3	88
2	8⅝ × 12	33.9	114.2	13.6	1.7	12.2	11.9	87.8
3	9½ × 13	25.8	116.1	13.8	2.2	16	11.6	84
4	8½ × 14	37.0	138.2	20.32	3.5	17.4	16.8	82.6
5	9 × 12	36.24	179.2	24.8	4.9	19.7	19.9	80.3
6	8½ × 12	31.25	168.8	18.0	3.6	20.3	14.4	79.7
Averages	—	—	140	17.6	2.9	16.3	14.7	83.7

One of the A compound portable engines of Messrs. R. Garrett & Sons, having 7-inch and 10½-inch cylinders, with 10 inches of stroke, on cranks at right angles, was tested by the brake, and yielded the following results:—

Resistance of Garrett's A Compound Portable Engine.

Speed in turns per minute................................	176.2	192.4
Indicator horse-power..............................I.H.P.	30.08	29.14
Horse-power by the brake..........................H.P.	26.13	25.65
Difference, frictional resistance....................H.P.	3.95	3.49
Do. per cent of the indicator power...........	13.1%	12.0%
Efficiency of transmission................................	86.9%	88.0%

The compound "undertype" engines constructed by Messrs. R. Hornsby & Sons, are placed horizontally, with inside cylinders under a boiler of the locomotive form, having cranks at right angles. In the following table, No. 135, are given particulars of these engines, of from 6 horse-power to 50 horse-power, nominal, with indicator horse-powers, resistances and efficiencies of transmission. The values of the resistances are evidently only approximate. But, taken at 16.7 per cent, they are almost identical with the average for single-cylinder portable engines, table No. 134.

The three horizontal engines—Corliss and Wheelock—already noticed, page 515, were tested for frictional resistance, as condensing and as non-condensing engines. The results deduced by the reporters are summarized in table No. 136; in which the friction of each engine running alone is distinguished from the additional friction due to the load. The power absorbed by the air-pump, including its frictional resistance, is stated separately; and the "net effective horse-power," or work given off at the brake on fly-wheel shaft, is the difference of the sum of all these resistances and the indicator power of the engine.

The Reynolds engine was fitted with an independent single-acting air-pump and jet-condenser, driven by a belt from the engine shaft. For the

Harris engine a double-acting air-pump and jet-condenser were used, driven from the crank-pin through a rocking lever. The Wheelock engine was provided with a Bulkley condenser, not requiring an air-pump, the air in the exhaust being carried down the descending limb of the condenser by induction. The average effective power, condensing, was 88.78 per cent; and non-condensing, 89.44 per cent. The difference in favour of the non-condensing action is .66 per cent, or less than 1 per cent.

Table No. 135.—RESISTANCE OF COMPOUND UNDERTYPE STEAM ENGINES, BY R. HORNSBY & SONS.

Nominal Horse-power.	Diameter of Cylinders.		Stroke.	Fly-wheel.		Speed in Turns per Minute.	Indicator Horse-power.	Horse-power at the Brake.	Difference.	Frictional Resistance.	Efficiency of Transmission.	Maximum Indicator Horse-power.
	1st Cylinder.	2d Cylinder.		Diameter.	Width.							
1	2	3	4	5	6	7	8	9	10	11	12	13
N. H. P.	ins.	ins.	ins.	ft. ins.	ins.	turns.	I. H. P.	H. P.	H. P.	per cent.	p. cent	I. H. P.
6	4⅛	8	10	4 0	5	220	18	15	3	16.7	83.3	20
8	5	9	12	4 6	6½	180	24	20	4	16.7	83.3	26
10	6	10½	14	5 0	8	155	30	25	5	16.7	83.3	33
12	6¼	11¼	14	5 0	8½	155	36	30	6	16.7	83.3	39
16	7¼	12¾	16	6 0	9	135	48	40	8	16.7	83.3	52
20	8	14	16	6 0	10½	135	60	50	10	16.7	83.3	65
25	9⅛	16	18	7 0	12	120	75	62	13	16.7	83.3	80
30	10	17½	18	7 0	14	120	90	75	15	16.7	83.3	96
40	12	21	24	1 wheel 8 0	20 12	90	120	100	20	16.7	83.3	128
				or	or							
50	13	23	24	2 wheels 7 0	25 15	90	150	125	25	16.7	83.3	160

NOTES TO TABLE.—1. The indicator power, column 8, is that which can be economically exerted.

2. The horse-power of engines fitted with condensers, is ⅛th to ¼th additional to what is given in columns 8, 9, and 13.

Table No. 136.—FRICTIONAL RESISTANCE OF CORLISS & WHEELOCK ENGINES.

Cylinders 18 inches in diameter, 4 feet stroke.

ENGINE.	Speed in Turns per minute.	Indicator Horse-power on Duty.	Frictional Resistance of Engine.					Net Effective Horse-power.	
			Alone.	Extra for Load.	Air-Pump.	Total Frictional Resistance.		Horse-power.	In Parts of I.H.P.
						Resistance.	In Parts of I H.P.		
CONDENSING.	turns.	I.H.P.	H.P.	H.P.	H.P.	H.P.	per cent.	H.P.	per cent.
Reynolds-Corliss	75.38	163.00	10.26	6.11	3.43	19.80	12.15	143.20	87.85
Harris-Corliss...........	75.83	165.58	9.57	6.24	4.69	20.50	12.38	145.08	87.62
Wheelock	74.47	158.38	7.81	6.02	0.60	14.43	9.12	143.95	90.88
NON-CONDENSING.									
Reynolds-Corliss........	75.33	137.02	10.26	5.07	—	15.33	11.69	121.69	88.61
Harris-Corliss...........	75.81	134.29	9.56	4.99	—	14.55	10.83	119.74	89.17
Wheelock	76.07	139.97	7.98	5.28	—	13.26	9.47	126.71	90.53

An ordinary Corliss engine in the spinning mill of Messrs. Schlumberger, Son, & Co., Mulhouse, was tested for resistance by MM. Walther-Meunier and G. Keller in 1878.[1] The cylinder was 2 feet in diameter, with a stroke of 4 feet. The power required to drive the engine in exerting various quantities of power, was measured by two friction-breaks applied to a shaft driven by toothed gearing from the fly-wheel. The speed was maintained at 50 turns, or 400 feet of piston per minute. The results of the tests for frictional resistance are given in table No. 137, first in order. The remarkable result is here exhibited, that the frictional resistance of the engine is virtually constant for increasing indicator powers.

A like constancy obtains in the results of tests of M. Hirn's single-cylinder engine already noticed, page 518, using saturated steam. The results of tests for frictional resistance were according to No. 2 in table No. 137; the speed being at the rate of 30 turns, or 336 feet of piston per minute.

M. Hallauer compared the frictional resistances of Corliss, Woolf, and compound horizontal engines,[2] with the results No. 3 in the table. In this, as in the preceding Corliss engine, No. 1 in the table, the frictional resistance is constant, for constant speed with varying power. But the Woolf and the compound engines show an increase of resistance with the indicator power. In all cases the efficiency of transmission increases with the indicator power.

In the case of a horizontal Woolf engine tested by M. Walther-Meunier,[3] a small increase of frictional resistance took place with the power. The test was made in three stages: 1st, when the brakes operated without a load—by their own weight only; 2d, when they were loaded up to the average work of the factory; 3d, loaded to the maximum power of the boilers. The normal speed of the engine was 39.34 turns, or 334.4 feet of piston per minute. The cylinders were 15 inches and 33¾ inches in diameter, with a stroke of 4 feet 3 inches. The results are No. 4 in the table. Here, the increase of absolute frictional resistance is only 3.55 horse-power, for an increase of 95.83 horse-power in the cylinders.

M. Hallauer, in 1876, tested a horizontal Woolf engine, having cylinders of 15 inches and 33½ inches diameter, with a stroke of 51 inches. Making 39.37 turns, or 167 feet of piston per minute, for 131.27 indicator horse-power, there was 113.35 horse-power at the break, being 86.3 per cent of the indicator power, and showing 13.7 per cent frictional resistance.

Another horizontal Woolf engine, running at a speed of 39½ turns per minute, indicating 130 and 181 horse-power, showed 112 and 161 horse-power, or 86 and 89 per cent at the brake, making the differences 18 and 20 horse-power, as frictional resistance, or 14 per cent and 11 per cent respectively.

The compound horizontal engine of Weyher & Richemond, noticed at page 544, when tested by the brake, yielded 87 per cent of power at the brake.

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[1] *Bulletin de la Société Industrielle de Mulhouse*, vol. xlviii. page 910. Also *Proceedings of the Institution of Civil Engineers*, vol. lviii. page 388.

[2] *Proceedings of the Institution of Civil Engineers*, vol. lxii. page 385.

[3] Ibid. vol. l. page 260.

Two Woolf beam engines, reported on by M. Hallauer,[1] one at Munster, and the other at Malmuspach, were tested for frictional resistance for various indicator powers at a speed of about 25 turns per minute. The first engine had cylinders 21.65 inches and 47¼ inches in diameter, with strokes respectively 4 feet 7 inches and 6 feet 7 inches in length. The results were according to No. 5 in the table following.

Table No. 137.—FRICTIONAL RESISTANCE OF SINGLE AND COMPOUND STEAM ENGINES:—SPEEDS CONSTANT, POWERS VARYING.

1. *Corliss Engine—Schlumberger & Co.*

Indicator horse-power	72.8	102.0	119.5	133.3	145.4	158.4
Horse-power absorbed by frictional resistance of engine and shaft	12.4	12.2	13.5	11.8	12.4	13.6
Do. do. per cent	17.0%	11.9%	11.6%	8.9%	8.6%	8.5%
Efficiency of transmission, per cent	83.0%	88.1%	88.4%	91.1%	91.4%	91.5%

2. *Hirn's Engine.*

Indicator horse-power H.P.	107	113	125	146	154
Horse-power by the brake H.P.	95	102	114	134	143
Difference, or frictional resistance H.P.	12	11	11	12	11
Do. do. per cent of the indicator horse-power	11%	10%	9%	8%	7%
Efficiency of transmission per cent	89%	90%	91%	92%	93%

3. *Corliss, Woolf, and Compound Horizontal Engines.*

	Corliss, One Cylinder.			Woolf, Beam.		Compound Horizontal (Marine).	
Indicator horse-power H.P.	105	137	158	150	220	64.5	78.5
Horse-power by the brake H.P.	92.4	123.3	145.4	123	189.2	55.5	68.3
Difference, or frictional resistance H.P.	12.6	13.7	12.6	27	30.8	9.0	10.2
Do. do. per cent of the indicator horse-power	12%	10%	8%	18%	14%	14%	13%
Efficiency of transmission	88%	90%	92%	82%	86%	86%	87%

4. *Horizontal Woolf Engine.*

Indicator horse-power H.P.	95.50	131.27	191.33
Horse-power by the brake H.P.	78.18	113.35	170.46
Difference, or frictional resistance H.P.	17.32	17.92	20.87
Do. do. per cent of the indicator power	18.1%	13.7%	10.9%
Efficiency of transmission	81.9%	86.3%	89.1%

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[1] *Bulletin de la Société Industrielle de Mulhouse*, 1878; page 232.

Table No. 137 (*continued*).

5. *Woolf Beam Engines.*

	Munster.			Malmuspach.				
Speed in turns per minute	25.4	25.2	25.2	26.2	25.9	24.2	24.8	25.5
Indicator horse-power, H.P.	185.75	267.85	347.16	143.11	149.53	201.64	212.92	215.7
Horse-power by the brake, H.P.	145.52	226	303.16	118.38	124.74	172.80	183.67	185.7
Difference, frictional resistance, H.P.	40.23	41.85	44.0	24.73	24.79	28.84	29.25	30.0
Do. do. per cent of the indicator power	21.7%	15.7%	12.7%	17.3%	16.6%	14.4%	13.8%	13.9%
Efficiency of transmission	78.3%	84.3%	87.3%	82.7%	83.4%	85.6%	86.2%	86.1%

From the results of tests of portable engines, table No. 134, page 612, it appears that the percentage of frictional resistance increases with the speed. But the data, being derived from engines of varying proportions and construction, cannot be very closely compared. It is remarkable that whilst the other single-cylinder engines have shown a practically constant absolute resistance for various indicator powers at equal speeds, the Woolf and other compound engines show absolute resistances increasing with the

Table No. 138.—SUMMARY RESISTANCE AND EFFICIENCY OF TRANSMISSION OF STATIONARY STEAM ENGINES.

ENGINE.	Cylinder: Diameter and Stroke.	Speed in Turns per Minute.	Indicator Horse-power.	Resistance.	Efficiency of Transmission.
	ins. ins.	turns.	I.H.P.	per cent.	per cent.
8 H.P. portables	9 × 12	114 to 180	13.6 to 24.8	13 to 22	87 to 78
A compound portable	7 × 10 10½ × 10	176 to 192	30 and 29	13 and 12	87 and 88
20 H.P. compound undertype	8 × 14	135	60	16.7	83.3
Corliss & Wheeler (three engines) average:					
Condensing	18 × 24	75	162	11.2	88.8
Non-condensing	" "	75	137	10.7	89.3
Corliss	24 × 48	50	73 to 158	17 to 8½	83 to 91½
Corliss	—	—	105 to 158	12 to 8	88 to 92
Hirn, beam	23¼ × 67	30	107 to 154	11.4	88.6
Compound horizontal	—	—	64½ to 78½	14 to 13	86 to 87
Compound horizontal (Weyher & Richemond)	11.18 × 18.9 18.9 × 18.9	—	—	13	87
Woolf horizontal	15 × 51 33¼ × 51	—	95.5 to 191	18 to 11	82 to 89
Woolf horizontal	15 × 51 33½ × 51	39.4	131.3	14.4	85.6
Woolf horizontal	—	39.5	130 and 181	14 and 11	86 and 89
Woolf beam	—	—	150 to 220	18 to 14	82 to 86
Woolf beam (Munster)	21.65 × 55 47.25 × 79	25.3	186 to 347	21.7 to 12.7	78.3 to 87.3
Woolf beam (Malmuspach)	—	25.3	143 to 216	17 to 14	83 to 86

indicator power. The drift of the evidence also leads to the conclusion that Woolf and compound receiver engines have a greater degree of frictional resistance than Corliss and other single engines. But the A compound portable is an exception. Its resistance, even at much higher speeds, does not exceed that of the single-cylinder portable engines. The leading results are given in table No. 138 preceding.

Pumping Engines.—The pumping engines of the East London Waterworks, already noticed (page 508), worked, when tested by Mr. Greaves, with the following frictional resistances, being the difference between the indicator power and the duty or work done in lifting water:—

Engine.	Cylinder.		Frictional Resistance.
Ajax	72-inch	......	6.64 per cent
Cornish	80 „	......	11.47 „
Wicksteed	90 „	......	9.51 „
Victoria	100 „	......	9.47 „
Average			9.23 per cent.

The 80-inch, 90-inch, and 100-inch engines made, on an average, six strokes per minute, with 140 horse-power of duty.

The Crossness beam pumping engines, already noticed (page 506), are double-acting single-cylinder engines, having 48-inch cylinders, with 9 feet of stroke, and a fly-wheel for each engine, beyond the pump. The pumps are single-acting, two to each engine, one on each side of the main-centre. The working speed of the engines is 11 turns, or 198 feet of piston, per minute. When the engines were tested, the lift was 20½ feet, and 202 indicator horse-power for each engine was developed. The duty, or net work done in lifting water, was 86 per cent, and the gross resistance was 14 per cent.

In the Woolf beam pumping engines, at the Lambeth Water-works, Thames Ditton, already noticed (page 533), the first cylinder is 28 inches by 5 feet 6⅜ inches stroke; the second is 46 inches by 8 feet. The speed of the large piston may be taken as 230 feet per minute, corresponding to 14⅜ turns per minute. The losses by friction of engines and pumps were from 12 per cent to 15 per cent of the indicator power; showing from 88 per cent to 85 per cent of useful work or duty, under a lift of 210 feet.

The receiver pumping engine (the J engine), already noticed, constructed by Messrs. James Simpson & Co., erected at the Lambeth Water-works, Ditton, has 21-inch and 36-inch cylinders, of 5½ feet stroke, making 24 revolutions per minute. Indicating 127.40 horse-power, the duty under a head of 35 feet, was proved by Mr. E. A. Cowper to amount to 77.4 per cent, showing 22.6 per cent of frictional resistance.

The Lawrence beam pumping engines, already noticed (page 538), have compound cylinders, 18 inches and 38 inches in diameter, with a stroke of 8 feet. Making 16.3 turns per minute, the duty was 83.7 per cent of the indicator power, and the total resistance 16.3 per cent.

Two Worthington direct-acting pumping engines, constructed by Messrs.

James Simpson & Co. for the West Middlesex Water-works, at Hampton, were tested in a 24-hours' trial by Professor W. C. Unwin. The two steam cylinders of each engine were 27 inches in diameter, with strokes of 4½ feet. The pump plungers were 40 inches in diameter. The engines made 17.28 double strokes per minute, developing for both engines 255.52 indicator horse-power. The duty in water lifted under a mean head of 60.63 feet, amounted to 85 per cent of the indicator power, with a resistance of 15 per cent.[1]

Another Worthington pumping engine, constructed by Messrs. James Simpson & Co., with compound cylinders, and compensating cylinders, at the New River Water-works, Stoke-Newington, was tested by Mr. E. A. Cowper. The first cylinders were 27 inches in diameter, and the second were 54 inches, with a stroke of 3.603 feet. The total number of strokes averaged 35.31 per minute, under a head of 148.48 feet of water, the pumps being 28 inches in diameter. The indicator horse-power was 330; the duty horse-power was 301.92, or 91.5 per cent of the indicator power, showing 8.5 per cent for frictional resistance.[2]

Summarizing these results of tests of pumping engines, they are as in the following table, No. 139:—

Table No. 139.—SUMMARY RESISTANCE AND EFFICIENCY OF TRANSMISSION OF PUMPING ENGINES

ENGINE.	Head of Water.	Speed in Double-strokes or Turns per Minute.	Resistance in parts of the Indicator H.P.	Efficiency of Transmission.
	feet.	double-strokes.	per cent.	per cent.
East London Water-works:—Cornish	140	6	9.23	90.77
Crossness Sewage Works:—Rotative, beam	20.5	11	14	86
Lambeth Water-works:—Rotative, Woolf-beam	210	14½	12 to 15	88 to 85
Lambeth Water-works:—Rotative, receiver-beam	35	24	22.6	77.4
Lawrence Water-works:—Rotative, compound beam	169	16.3	16.3	83.7
West Middlesex Water-works:—Worthington	60.63	17¼	15	85
New River Water-works, Worthington	148.48	35.31	8.5	91.5

DISTRIBUTION OF FRICTIONAL RESISTANCE IN STEAM ENGINES.

Professor Thurston has made a long series of trials with four types of steam engines, to determine how much power is lost in friction at the different working parts:—1. A "straight-line" engine, having a 6-inch cylinder, with a 12-inch stroke, with a balanced valve and an unbalanced valve. 2. A traction engine, having a 7-inch cylinder, with a 10-inch stroke, and locomotive valve gear. 3. An engine having a 12-inch cylinder, with 18 inches of stroke, and an automatic balanced expansion valve. 4. A condensing engine, having a 21-inch cylinder, with a 20-inch stroke, and a balanced valve. From a paper on the subject read at the meeting of the American Society of Mechanical Engineers, in October, 1888, the following table is derived:—

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[1] These particulars are derived from the report as published in *Engineering*, December 7, 1888, page 565.

[2] See *Engineering*, January 4, 1889, page 10.

Table No. 140.—Distribution of Friction in some Direct-acting Steam Engines. Dr. Thurston.

Parts of Engine.	Percentages of Total Friction.				
	"Straight Line" 6 in. by 12 in. Balanced Valve.	"Straight Line" 6 in. by 12 in. Unbalanced Valve.	7 in. by 10 in. Lansing Iron-works—Traction: Locomotive Valve Gear.	12 in. by 18 in. Lansing Iron-works—Automatic, Balanced Valve.	21 in. by 20 in. Lansing Iron-works—Condensing, Balanced Valve.
Main bearings	47.0	35.4	35.0	41.6	46.0
Piston and rod	32.9	25.0	21.0	49.1	21.8
Crank pin	6.8	5.1	13.0		
Cross head and wrist pin	5.4	4.1			
Valve and rod	2.5	26.4	22.0	9.3	21.0
Eccentric strap	5.3	4.0			
Link and eccentric	—	—	9.0	—	—
Air pump	—	—	—	—	12.0
Total	100.0	100.0	100.0	100.0	100.0

It may be remarked, in conclusion, on the friction of steam engines, that the degree of nearness to uniformity of frictional resistance for various powers of the same engine, at the same speed, is probably dependent upon the degree of nearness by which the momentum of the reciprocating parts is balanced by the pressure of the steam.

Addendum.—A test-trial was recently (March, 1889) made by the author of a small non-condensing steam engine, vertical, having the cylinder overhead, working to the crank shaft below, with an ordinary slide valve worked by an eccentric. The cylinder was 4 inches in diameter, with a stroke of 5 inches, and steam was cut off at one-fourth of the stroke, at a pressure of about 29 lbs. per square inch. At an average speed of 197 turns per minute, the engine indicated .780 horse-power; and the brake on the fly-wheel .337 horse-power, or only 43.2 per cent of the indicator power. This result is confirmatory of the statement already made, that the smaller the engine, the greater is the friction proportionally.

SECTION III.

THE CONSTRUCTION OF STEAM BOILERS.

CHAPTER I.—ELEMENTAL STRENGTH OF STEAM BOILERS.

I. STRENGTH OF A CYLINDRICAL SHELL.

The total pressure on a 1-inch length of both sides together of a cylindrical boiler is equal to the product of the internal diameter by the pressure of steam per square inch. Let—

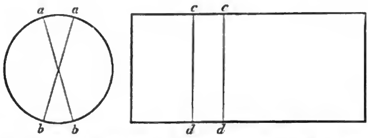
d = the internal diameter, in inches.
t = the thickness of metal at each side, in inches.
s = the ultimate tensile or breaking strength of the metal, in tons per square inch.
p = the steam-pressure, in pounds per square inch.

Then $d \times p$ is the total pressure on a 1-inch length of both sides together; $2t$ is the sectional area of both sides; and $2ts \times 2240 = dp$; or,

$$p = \frac{4480\,ts}{d}; \quad t = \frac{dp}{4480\,s}; \text{ and } s = \frac{dp}{4480\,t} \ldots\ldots\ldots (1), (2), (3)$$

For riveted joints, the tensile strength, s, is to be taken as the product of that of the whole plate by the coefficient of strength.

Longitudinal Pressure.—When the ends of a cylinder are closed, and make one piece with the cylindrical portion, the total longitudinal resistance to internal pressure is directly proportional to the thickness of the cylindrical portion, and to the diameter; whilst the total bursting force acting on the ends is proportional to the area of one end, or to the square of the diameter. The bursting pressure per square inch that can be resisted longitudinally as well as transversely is inversely proportional to the diameter. Let the circle and the rectangle, figs. 257, be a cross section and a longitudinal section of a cylindrical boiler. The area of the circle is a measure of



Figs. 257.—Diagram for the Resistance of a Flat-headed Cylinder to Bursting Pressure.

the longitudinal pressure of the steam on the ends of the boiler. Set off the interval *a a* on the circular section, and the equal interval *c c* on the longitudinal section; and draw the diameters *ab*, *ab*, and the parallels *cd*, *cd*. The areas of pressure to be resisted are respectively the two triangular spaces *ab*, and the rectangle *cd*; and, since the former have only half the surface of the latter, it follows that the longitudinal stress per square inch on the shell is only half the transverse stress.

On the same showing, a hollow sphere can resist twice the pressure per square inch that a tube of equal diameter and equal thickness can do.

II. STRENGTH OF FLAT ENDS AND SEGMENTAL ENDS OF BOILERS AND OTHER CYLINDERS.

The end of a cylindrical boiler, consisting of a perfectly flat unstayed plate of uniform thickness, springs and bulges outwards when it is subjected to internal pressure. The form of the cavity is spheroidal rather than spherical:—the deflection or bulging being greatest at the centre. The bulging is accompanied by an extension of the plate radially, under the tensile stress which is set up; and it may appear, at first sight, that the bulged form is a segment of a sphere. But, according to the form of a sphere, the tensile stress must be exerted equally in every direction, that the extension may take place circumferentially as well as radially, and at the same rate. This is the essential condition of true spherical expansion. Whereas, in the bulged end, extension of the plate circumferentially is resisted by the cylindrical body of the boiler, which acts as a curb; and the internal stress of radial extension is accompanied by circumferential compression. The form of the cavity is, therefore, more nearly elliptical than spherical, the shortest radius of curvature being at the circumference. Nevertheless, the curvature of the bulge, within the very limited range in practice, is so nearly identical with that of a sphere, that it may, for the purpose of calculation of strength, be assumed to be spherical. The inference is that the line of ultimate fracture is not diametral, or in the line of a great circle of the sphere. It is circumferential, that is, at right angles to the radius of the cylinder, at the outer circumference, in the line of the fastening,—where failures in fact do take place.

The stresses in the end-plate,—the radial-tensional and the circumferential-compressive,—are balanced by longitudinal stress in the body of the boiler. The relations of these stresses, and formulas deduced therefrom, have been fully treated by the author.[1]

The conclusions and formulas are here quoted:—

Ultimate Elastic Deflection of End-plate.

$$\delta = \frac{\text{radius}}{22} = \frac{r}{22} \ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

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[1] See a paper "On the Strength of Flat Plates and Segmental Ends of Boilers and other Cylinders," in the *Minutes of Proceedings of the Institution of Civil Engineers*, volume liii. 1878; page 170.

$$\delta = \frac{\text{diameter}}{44} = \frac{d}{44} \quad \ldots\ldots \quad (5)$$

δ = the deflection in inches.
r = the radius in inches.
d = the diameter in inches.

Allowing that the elastic limit is reached under a strain of $^1/_{1000}$th of the diameter of the cylinder, representing a stress of 12 tons per square inch, the diameter and the radius of the spherical bulging are given by the formulas—

Diameter and Radius of the Segmental Sphere at the Elastic Limit.

$$D = 22\,r \quad \ldots\ldots \quad (6)$$

$$D = 11\,d \quad \ldots\ldots \quad (7)$$

$$r' = 11\,r = 5.5\,d \quad \ldots\ldots \quad (8)$$

D = the diameter of the sphere, in inches.
d = the diameter of the circular junction of the end-plate with the cylinder.
r' = the radius of the sphere.

A hollow sphere, like a hollow cylinder strained longitudinally, is capable of resisting twice the bursting pressure per square inch that would rupture a hollow cylinder transversely of the same diameter. The formula for the bursting strength of a hollow cylinder may be adapted for calculating the resistance of a sphere to internal pressure by doubling the coefficient; when it becomes—

$$p = \frac{4480\,t\,s \times 2}{D} = \frac{8960\,t\,s}{D} \quad \ldots\ldots \quad (9)$$

p = internal pressure, in lbs. per square inch.
t = thickness of plate, in inches.
s = tensile strength of the plate, in tons per square inch.
D = diameter of the sphere, in inches.

This formula may be adapted directly for a flat end-plate, strained by internal pressure to the elastic limit, by substituting for D its value when the radial extension of the plate is $^1/_{1000}$th, namely $11\,d$—

Relative Internal Pressure and Stress in the End-plate strained to the elastic limit.

$$p = \frac{8960\,t\,s}{11\,d} = \frac{815\,t\,s}{d} \quad \ldots\ldots \quad (10)$$

p = internal pressure, in lbs. per square inch.
t = thickness of the plate, in inches.
s = tensile stress in the plate, in tons per square inch at the elastic limit.

This equation is good for steel plates, as well as for iron plates, since the elastic limit of strain, taken generally, is the same for both metals, namely, $^1/_{1000}$th of the length. The values of s, the elastic strengths, are—

For iron, 12 tons per square inch.
For steel, 14 " "

Substitute these values for s in the formula (10), and the following formulas are obtained, in which the coefficients have been reduced to round numbers—

Elastic Strength of Circular flat End-plates of Iron and of Steel, of uniform thickness, fastened at the circumference, exposed to bulging pressure uniformly distributed.

$$\text{For iron} \ldots\ldots p = 10000 \frac{t}{d} \ldots\ldots \quad (11)$$

$$\text{For steel} \ldots\ldots p = 11500 \frac{t}{d} \ldots\ldots \quad (12)$$

p = bulging pressure, in lbs. per square inch.
t = thickness of the plate, in inches.
d = diameter of the plate, measured to the circular line of junction.

Since the end-plate is thrown by deflection into a state of tension throughout its thickness, the resistance to bursting pressure follows the ratio of the thickness simply: not the ratio of the square of the thickness, as is assumed for the most part by writers on the resistance of flat surfaces. The experiments of Sir William Fairbairn on the resistance of ¼-inch and ½-inch plates to bulging stress, applied at the centre, showed clearly that the resistance increased directly as the thickness. The experiments of Mr. Kirkaldy point to the same conclusion.[1]

The magnitude of the second stress—the circumferential-compressive stress—is found by means of the formula[2]—

The Compressive Stress at the Circumferential Junction of a Flat End-plate.

$$c = \frac{p \times r \times \sqrt{r'^2 - r^2}}{2} \ldots\ldots \quad (13)$$

c = total compressive stress at the circular line of junction, in pounds.
p = internal pressure, in pounds per square inch.
r = internal radius of the cylinder, in inches.
r' = radius of the spherical end, in inches.

Though the compressive stress has been supposed, for the sake of illustration, to be concentrated in the line of junction between the end-plate and the cylinder, it is actually distributed over the whole of the diametral section, from side to side of the end-plate. But the distribution is not uniform. The stress is a maximum at the circular line of junction, and it obviously vanishes at the axial line, where the bulging of the end-plate, or deviation from a plane, is equal to nothing.

For the case in which the metal—iron or steel—of the flat end-plate, is strained radially to the limit of its elastic strength, this formula may be used:—

Compressive Stress at the Circumferential Junction of a Flat End-plate when the radial strain reaches the elastic limit.

$$c = \frac{p \times r \times 11\, r}{2}; \text{ or}$$

$$c = 5.5\, p\, r^2 \ldots\ldots \quad (14)$$

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[1] See *A Manual of Rules, Tables, and Data*, page 569.

[2] This formula is the same in construction as that given by Rankine in *A Manual of Civil Engineering*, 1863; page 228.

Example.—The flat end-plate of a cylindrical wrought-iron steam boiler is ½ inch thick, and is 24 inches in diameter, measured to the centre line of the circle of rivets forming the joint.

The elastic deflection at the centre, by formula (5), in which $d=24$, is equal to $^{24}/_{44}=.55$ inch. The elastic bulging pressure by formula (11), for which $t=\frac{1}{2}$, is equal to $p=\frac{10000\times\frac{1}{2}}{24}=208$ lbs. per square inch. The compressive stress at the circumference is by formula (14),

$$c=5.5pr^2=5.5\times 208 \text{ lbs.}\times 12^2=164{,}736 \text{ lbs., or } 73.5 \text{ tons.}$$

It has been found that the elastic resistance of wrought iron, when in quantity, to compression, may be taken at 20 tons per square inch. An area of $\left(\frac{73.5}{20}=\right)$ 3.675 square inches, would therefore have elastic resistance sufficient to oppose the compressive stress in this example.

To compare the elastic resistance of this flat end-plate with that of the cylindrical portion of the boiler to which it is attached, 24 inches in diameter and of ½-inch plate. By the usual formula (1),

$$p=\frac{4480\,ts}{d}=\frac{4480\times\frac{1}{2}\times 12}{24}=1120 \text{ lbs. per square inch,}$$

reckoned for solid plate. This value is 5.4 times the value for the elastic strength of the flat end-plate, which is 208 lbs. per square inch. Passing over, meantime, the reduction of strength at the joints, which would have the same proportion for both the parts, it would follow that a flat end-plate, to be equally strong elastically with the cylindrical portion, would require to have 5.4 times the thickness of the other, which would amount to $(\frac{1}{2}\times 5.4=)$ 2.7 inches. For ⅜-inch body-plates, the required thickness for the end-plate would be $(\frac{3}{8}\times 5.4=)$ 2 inches.

The ratio of resistance for flat unstayed end-plates and cylindrical bodies of equal thicknesses, namely 1 to 5.4, is the same for boilers of all diameters, supposing that the line of junction is at the edge of the cylinder.

In these analyses of stress and strain, the effect of riveted joints in reducing the strength has not been considered. Inasmuch as the weakening influence of riveted joints is the same proportionally for end-plates as for cylindrical-plates, the ratios of strength remain unaffected by the joints, although the absolute values are reduced by them. But the reduction of strength at the joint does not necessarily affect the play of the end-plate within the elastic limits of the solid plates; although it may induce a slight degree of permanent set, which, as Mr. Robert Wilson explains,[1] may be advantageous, "in preventing the plate from becoming grooved by the continual variation of pressure in working," and the consequently varying deflection.

A diametrical riveted joint across a flat end-plate does not necessarily reduce the resistance of the plate to bulging stress, since the tensional stress is radial, and is therefore coincident with the seam.

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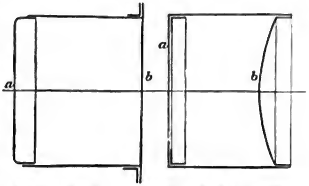
[1] "The Strength of Unstayed Flat Surfaces," *Engineering*, September 28, 1877; page 240.

Mr. S. Nicholls[1] gives the results of four experiments made by him in 1876, partly under the direction of Mr. Wilson, for testing the strength of unstayed flat ends of boilers.

A sample piece, fig. 258, was constructed by Mr. Nicholls, 2 feet 6 inches in diameter inside, and 2 feet long. The body was made of one 7/16-inch iron plate, butt-jointed, with a double-riveted single welt. One end *a* was closed with a ⅜-inch flat plate, formed with a 2½-inch flange turned to a radius of 1 inch inside, and riveted to the body. The other end *b* was closed with a 9/16-inch plate riveted to an angle-iron ring, 3 inches by ½ inch thick, on the outside of the body. The extreme diameter outside of the ring was 3 feet ⅞ inch, and the end-plate was joined to the angle-iron by a line of ⅞-inch rivets at a pitch of 2¼ inches, of which the centre-line formed a circle 2 feet 10½ inches in diameter. The rivet-holes were punched. When tested by hydraulic pressure, the 9/16-inch plate failed in the line of the rivet-holes. For the second trial, a ⅜-inch plate was substituted for the 9/16-inch plate which had failed. When proved, it failed in the same way—across the rivet-holes. For the third trial, an inverted dished plate, ⅜ inch thick, was substituted, as in fig. 259, for the ⅜-inch flat plate which failed in the second trial. This plate was dished to a radius of 3 feet, and was formed with a 3-inch flange; it was inserted with the convex surface inwards, and riveted direct to the body. The pressure-gauge got out of order, and the trial was abandoned. For the fourth trial, the ends were cut out and a 3-inch angle-iron, ½ inch thick, was inserted at *a*, fig. 259, inside the body, to which a ⅜-inch flat plate was riveted, with a line of ¾-inch rivets at a pitch of 2 inches, in a circle 2 feet 2¼ inches in diameter. The plate failed by fracture through the rivet-holes. For the fifth trial, the ⅜-inch flanged plate *a*, fig. 258, originally applied, and then removed, was again inserted and riveted through the flange to the body. This plate failed chiefly at three places in the turn of the bend, on the inner side, measuring 28½ inches apart diametrically.



Figs. 258 and 259.—Experimental Pieces for End Pressure. Scale 1/24th.

Although the failures through rivet-holes, in the first, second, and fourth trials, only resulted in complete fracture for one or two short lengths in each specimen,—through rimed rivet-holes in the first and second trials,—nearly every rivet-hole was damaged more or less, and when complete fracture did take place, the pressure instantly subsided. But the flanged plate behaved otherwise. It failed by two stages; after the first failure, when considerable leakage took place, the pressure was still further increased before final rupture was effected. The inverted segmental plate which was riveted to one end of the boiler, and employed as an abutment for the bursting pressure, resisted without set the highest pressures applied. Its maximum deflection was $^1/_{16}$ inch. The principal results are here abstracted:—

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[1] *The Theoretical and Practical Boiler Maker*, 1876; page 141.

No.	Thickness of Plate.	Diameter of Line of Fastening.	Deflection at 120 lbs. Pressure.	Bursting Pressure.	Final Deflection.
1	$^9/_{16}$ in.	2 ft. 10½ ins.	⅝ in.	280 lbs.	1¾ ins.
2	⅜ ,,	2 ,, 10 ,,	—	200 ,,	—
4	⅜ ,,	2 ,, 2¼ ,,	⅜ in.	370 ,,	1⅛ ins.
5	⅜ ,,	2 ,, 4½ ,,	$^{11}/_{16}$ in.	300 to 380 lbs.	1¾ ,,

Deflection by bulging commenced with the application of pressure, and the increase of deflection was practically uniform until about one-half the bursting pressure was applied. For higher pressures, the deflection increased more slowly; the decreased rate of increase was clearly due to the increasing convexity of the plate and the decreasing diameter of the sphere of convexity. The plates took a set early in the course of proof, and it is likely that the set arose from a yielding in the body of the plate. Calculating the deflection and the pressure within the elastic limits, by formulas (5) and (11)—

No.	Elastic Deflection.	Elastic Pressure.
1	$\frac{34.5}{44} = .78$ inch.	$10{,}000 \times \frac{.5625}{34.5} = 163$ lbs. per square inch.
2	$\frac{34.5}{44} = .78$,,	$10{,}000 \times \frac{.375}{34.5} = 109$,, ,,
4	$\frac{26.25}{44} = .60$,,	$10{,}000 \times \frac{.375}{26.25} = 143$,, ,,
5	$\frac{28.5}{44} = .65$,,	$10{,}000 \times \frac{.375}{28.5} = 132$,, ,,

The calculated elastic pressures range well within the proved bursting pressures, and the elastic deflections for the fourth and fifth experiments are in harmony with the experimental observations of deflection. But the elastic pressure for No. 1 is relatively high for the bursting pressure, whilst the elastic deflection is much lower than the experimental deflection for that pressure. These two comparisons for No. 1 point to the existence of a partial crippling of No. 1 plate, which, for its greater thickness, was no doubt affected to a greater degree by the bending stress at the joint, due to lap-riveting, than the ⅜-inch plates. The flanged plate gave evidence of circumferential compressive action in the contracting of the circle, and the widening of the angle at the bend; whilst, by the formation of the square angle for the flange, an element of contrary flexure near the circumference was introduced, which aided in reducing the diameter of the spherical surface in tension.

Cast-iron Ends.—Although there is no definite elastic limit for cast iron, short of the ultimate strength, it is convenient to adopt an extension of $^1/_{1000}$ part of the length, the same as for iron and steel, as the working elastic limit, corresponding to a tensile stress of 5 tons per square inch for ordinary castings. The formula for elastic deflection will be the same as that for iron or steel. For the elastic bulging pressure, the coefficient in formula (11) for iron, is to be reduced in the ratio of 12 tons to 5 tons; then $(10{,}000 \times \frac{5}{12} =)$ 4166 is the coefficient for cast iron; say 4000.

Elastic Strength of simple Flat Cast-iron Ends of Cylinders.—Iron of ordinary strength.

$$\delta = \frac{d}{44} \qquad (15)$$

$$p = 4000 \frac{t}{d} \qquad (16)$$

δ = deflection at the centre, within the elastic limit, in inches.
d = diameter of the line of fastening, in inches.
t = thickness of the plate, in inches.
p = elastic bulging pressure, in lbs. per square inch, uniformly distributed.

For cast iron of stronger quality, the numerical coefficient in formula (16) is to be increased in proportion.

Upon the whole, the results of Mr. Nicholl's experiments afford satisfactory evidence of the validity of the formulas for the elastic deflection and the elastic strength of simple flat ends for cylindrical vessels of iron or of steel, subjected to internal uniform pressure.

Segmental Ends.—The resistance of segmental ends of cylinders is calculated on the same principles as that of flat ends.

Diameter of the Segmental Sphere.

$$D = \frac{r^2}{v} + v \qquad (17)$$

r = radius ($a\,d$) of the circular junction, in inches, or of the cylinder when not otherwise described.
d = diameter ($a\,c$) of the circular junction, in inches.
v = versed sine ($d\,d'$), or the rise of the segment, in inches.
D = diameter of the sphere, in inches.
p = internal pressure, in lbs. per square inch.
t = thickness of the plate, in inches.
s = tensile stress in the plate, in tons per square inch.

The internal pressure due to the value of D, the diameter of the sphere, is given by the formula (9), here repeated—

$$p = \frac{8960\, t\, s}{D} \qquad (18)$$

in which is to be substituted the value of D above given:—

Relative Internal Pressure and Stress in the Segmental End.

$$p = \frac{8960\, t\, s}{\frac{r^2}{v} + v} \qquad (19)$$

Substitute the values of s: 12 tons for wrought iron, 14 tons for steel, and 5 tons for cast iron. Reducing, the formula becomes, for a whole plate, taking the resulting coefficients in round numbers:—

Elastic Strength of Segmental or Spherical Ends of Cylinders of uniform thickness.

$$\text{Wrought iron, } p = \frac{108000\,t}{\frac{r^2}{v}+v} \quad \ldots\ldots (20)$$

$$\text{Steel, } p = \frac{125000\,t}{\frac{r^2}{v}+v} \quad \ldots\ldots (21)$$

$$\text{Cast iron, } p = \frac{45000\,t}{\frac{r^2}{v}+v} \quad \ldots\ldots (22)$$

Example.—A cylindrical iron boiler, 2 feet in diameter, has a spherical segmental end, of which the rise or versed sine is 6 inches. The plates are ½ inch thick. What is the elastic strength of the end of the boiler? Using formula (20), $t = \frac{1}{2}$, $r = 12$, and $v = 6$; then $\frac{r^2}{v} + v = \frac{12^2}{6} + 6 = 30$ inches, the spherical diameter; and $p = \frac{108000 \times \frac{1}{2}}{30} = 1800$ lbs. per square inch, the elastic pressure per square inch for the end. To compare this value with the elastic pressure for the body of the boiler:—By the ordinary formula (1), page 620, $p = \frac{4480 \times \frac{1}{2} \times 12}{24} = 1120$ lbs. per square inch for the whole plate.

The allowance that should be made for the comparative weakness of a riveted joint at the circumference of a segment is an uncertain element. Supposing for the present that the strength of the end-plate is reduced in the same proportion as that of the body, it would follow that the spherical end is of the same elastic strength as the body, measured by the internal pressure in lbs. per square inch, when the spherical diameter is twice the diameter of the body, the thickness of plate being the same. In the present example the spherical diameter would be $(24 \times 2) = 48$ inches, instead of 30 inches, for equivalent strengths.

To find the proportional rise or versed sine of the segment of equal elastic strength for equal thickness of plate; the diameter of the sphere, $\left(\frac{r^2}{v}+v\right)$, is equal to twice the diameter, or four times the radius, $4r$; or $\left(\frac{r^2}{v}+v\right) = 4r$. Reducing for the value of v,

Versed Sine of Segmental End of equal strength.

$$v = 0.27\,r \quad \ldots\ldots (23)$$

that is to say, the versed sine or rise at the centre of a spherical segment, having the same thickness and elastic strength as the body of the cylinder, is about one-fourth of the radius of the end of the cylinder, or one-eighth of the diameter.

For a diameter of 6 feet, for example, the rise or dish of the spherical end of equal strength would be 9¾ inches.

The tensional stress in the metal due to a given pressure and given dimensions is found by an inversion of formula (19):—

Tensile Stress in the Segmental End for a given pressure.

$$s = \frac{p\left(\frac{r^2}{v} + v\right)}{8960\,t} \quad \text{(24)}$$

III. STRENGTH OF STAYED FLAT SURFACES OF BOILERS.[1]

The flat plate-surfaces of steam boilers are stayed by rows of bolts, pitched, usually, at equal intervals, vertically and transversely, dividing the surface into rectangles or squares. When internal pressure is applied on one face of the plate, it is bulged, and resolves itself into a group of circular or arched segments starting from the stay-bolts, in the manner represented by a tufted cushion. The segments are mitred together like groined arches, and they merge in the elevated interspaces between the stay-bolts. In section, the plate is formed as a series of arches from bolt to bolt. Like a flat end-plate, the stayed-plate is qualified for resisting internal pressure, by assuming a curved form. But whilst the flat end-plate takes the form of a spherical segment, the radical form of the stayed-plate when deflected is cylindrical, or at least approximately cylindrical; and its capability for resistance can be calculated accordingly.

When the plate is bulged to the elastic limit of extension, the deflection or rise for the span, or clear space between two stay-bolts, may be calculated by the formula (5), page 622, here repeated.

Rise or Deflection of Flat Stayed-plates to the Elastic Limit.

$$\delta = \frac{d}{44} \quad \text{(25)}$$

δ = rise of the arched plate, in inches.
d = clear distance apart between the two stay-bolts.

When the metal—iron or steel—is strained to the elastic limit, or $^1/_{1000}$th of the span d between two bolts, the diameter D of the segmental circle is, by formula (7), page 622.

$$D = 11d \quad \text{(26)}$$

and by formula (1), the corresponding pressure is $p = \frac{4480\,t\,s}{D}$; or it is, by substituting for D its value, 11 d, and reducing,

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[1] This subject has already been investigated by the author in his paper named in foot-note, page 621.

Relative Internal Pressure and Stress in a Flat Stayed-plate, strained to the Elastic Limit.

$$p = \frac{407\,t\,s}{d} \qquad (27)$$

p = internal pressure in lbs. per square inch.
t = thickness of the plate in inches.
d = clear maximum distance apart between two stay-bolts.
s = tensile stress in the plate, in tons per square inch, at the elastic limit.

The formula (27) may be reduced for iron, for steel, and for copper plates by substituting their values of s, 12 tons, 14 tons, and 8 tons respectively, and reducing the coefficients to round numbers:—

Elastic Strength of Flat Stayed-plates, of uniform thickness, exposed to bulging pressure uniformly distributed.

For iron......... $p = 5000\frac{t}{d}$ .... (28)

For steel .... $p = 5700\frac{t}{d}$ .... (29)

For copper...... $p = 3300\frac{t}{d}$ .... (30)

Thickness of Plate, and Distance apart of the Stay-bolts.

For iron......... $t = \frac{p\,d}{5000}$ .... (31)

" .... $d = \frac{5000\,t}{p}$ .... (32)

For steel .... $t = \frac{p\,d}{5700}$ .... (33)

" .... $d = \frac{5700\,t}{p}$ .... (34)

For copper...... $t = \frac{p\,d}{3300}$ .... (35)

" .... $d = \frac{3300\,t}{p}$ .... (36)

When the pitches of the stay-bolts, vertically and transversely, are unequal, the pitch of the greater length is to be adopted in the foregoing formulas.

The stay-bolts should be so proportioned that they may not be stretched beyond elastic limits, when the plate is subject to the maximum elastic pressure. Stress is applied to them in two forms—tensile stress and shearing stress.

With regard, first, to tensile stress, the product of the net sectional area of the bolt by the elastic tensile strength per square inch, should be at least equal to the total elastic pressure which the plate is capable of resisting. That is, on this assumption, $.7854 d'^2 s = \frac{\mathrm{P} \times \mathrm{P}' \times p}{2240}$; and $d'^2 = \frac{\mathrm{P} \times \mathrm{P}' \times p}{.7854 \times 2240 \times s}$

$=\frac{P\,P'p}{1760\,s}$; whence, taking the square root of both sides, $d=\sqrt{\frac{P\,P'p}{1760\,s}}$; or

Diameter of Stay-bolts (at base of thread) Strained to the Elastic Limit.

$$d'=.0024\sqrt{\frac{P\,P'p}{s}} \;\ldots\ldots\ldots\ldots\ldots\ldots \;(37)$$

d' = diameter of the stay-bolts, at the base of the thread.
P = pitch of the stay-bolts between centres, in inches longitudinally.
P′ = pitch of the stay-bolts between the centres, in inches, transversely.
p = total elastic pressure, in lbs. per square inch, on the plate.
s = elastic tensile strength, in tons per square inch.

For bolts of iron, steel, or copper, having respectively 12 tons, 14 tons, and, say, 8 tons per square inch, elastic tensile strength, the formula becomes, by substitution of these values for *s*, and reducing—

Proper Diameter of Stay-bolts (at base of thread) for Flat Stayed-plates.

When P = P′

Iron ......$d'=.00069\sqrt{P\,P'p}$; or $d'=.00069\,P\sqrt{p}$ .... (38)
Steel......$d'=.00064\sqrt{P\,P'p}$; or $d'=.00064\,P\sqrt{p}$ .... (39)
Copper...$d=.00084\sqrt{P\,P'p}$: or $d'=.00084\,P\sqrt{p}$ .... (40)

With regard, next, to the amount of longitudinal shearing stress on the bolt, it may be taken practically as identical with the shearing stress on the plate, although, in reality, the area for shearing stress in the plate is somewhat more than that in the bolt. The shearing resistance round each stay-bolt is usually evaded, where stay-bolts are merely screwed into the plate, by the drawing of the plate off the bolt. The same form of rupture takes place even if the ends of the bolts be riveted over on the plate:—the head is snapped off, or sheared off, and then the plate is drawn off the bolt. Occasionally, when the bolts are closely pitched, the screw-thread is stripped, either from the plate or from the bolt; but most commonly the plate becomes so bulged and stretched by pressure, and the hole so widened, that the plate slips off the bolt without materially damaging the threads. The actual quantity of this resistance, which may be classed as shearing resistance, is, therefore, in general, much less than that which would be offered by a perfectly fitting screwed bolt fairly strained. If the screwed joint, between the plate and the bolt, were a perfectly close fit, the shearing resistance at the screw would be measured by the area of the screwed surfaces in contact, measured at the base of the thread; and would be equal to the product of this area in square inches by the shearing resistance per square inch. The shearing resistance of iron may be taken as 80 per cent of the direct tensile resistance, but so large a proportion cannot be rendered at the screwed surfaces of the plate and the bolt. Mr. Brunel found that the nut, ¾ inch deep, of a screwed bolt, 1¼ inches in diameter in the shank, could strip the thread from the bolt, but that a ⅞-inch nut could not do so. Here, the area of equivalent shearing resistance was 2⅓ times the sectional area of the bolt; although, had the full measure of shearing resistance—80 per cent of the tensile resistance per square inch—been exercised, the shearing area need not have exceeded $\left(\frac{100}{80}=\right)$ 1¼ times the sectional area. In this case, the effective shearing resistance per square inch amounted scarcely to half the tensile resistance. In another case—that of a stay-bolt screwed into a plate, the result of an experiment made by Sir Wm. Fairbairn—the effective shearing resistance per square inch amounted to one-half of the tensile resistance:—a larger proportion than that which was effected by the nut, and reasonably larger, as the nut was of narrow lateral dimensions.

Under the most favourable circumstances, when the stress is fairly applied and bulging movement is prevented, the maximum effective resistance per square inch of the shearing surface of a screwed bolt in a plate may be taken as only 50 per cent of the tensile resistance of the bolt per square inch of section. In order that the measure of shearing resistance of a screwed stay-bolt should be equal to that of its tensile resistance, therefore, the area of shearing resistance should be twice the sectional area of the bolt, as is expressed by the equation $(d' \times t \times 3.14) = (d'^2 \times .7854 \times 2)$, in which d' is the diameter of the bolt at the base of the thread, and t the thickness of the plate. Reducing for d',

$$d' = 2\,t \ldots\ldots\ldots\ldots\ldots\ldots \quad (a)$$

showing that the equality of shearing resistance and tensile resistance of an iron stay-bolt would be attained when the net diameter of the bolt is twice the thickness of the plate, on the condition that the stress is fairly applied, without injurious distortion of the plate. But this is a condition which is not usually fulfilled in practice; and, even if the conditions of simple stress were fulfilled, it may be seen that the two formulas for the value of the diameter d' of the stay-bolt are not commensurable; for whilst the first value, by formula (34), varies as the square root of the pressure, the second, by equation (a), varies as the thickness of the plate, and therefore also as the pressure simply. It is the better course to select the first of these formulas, (34), for determining the size of the bolt proper for resisting tensile stress, and to make up, by way of compensation, if necessary, the needful power of resistance to rupture between the bolt and the plate.

There are three means of maintaining, wholly or in part, the resisting capacity of stay-bolts. First, by limiting the extent of the pitch of the bolts, so that the loosening effect of bulging strain may be minimized; second, by riveting over the ends of the stay-bolts upon the plates; third, by extending the ends of the stay-bolts to receive nuts which are to be screwed with washers on the plates. Riveting, which must be done when the bolt is cold, and therefore done only to a limited extent, is satisfactorily effective only when the stay-bolts are placed at short pitches of about 4 inches. But the addition of a nut at each end supplies an efficient means of ensuring that the resistance of the stay-bolt in its place in a boiler shall be brought up to the full measure of its tensile strength.

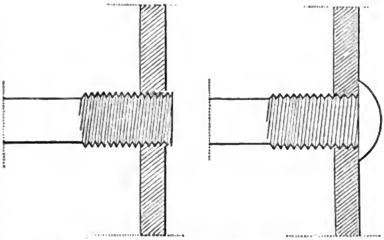
Example.—A flat iron plate, ½ inch thick, is stayed by rows of 1⅛-inch bolts placed rectangularly at a pitch of 12 inches both ways. The clear distance apart of the stay-bolts is (12 − 1⅛ =) 10⅞ inches. The deflection at the elastic limit is, by formula (25), equal to $\frac{d}{44} = \frac{10.875}{44} = \frac{1}{4}$ inch; and the elastic limit of pressure is, by formula (28), equal to $5000 \times \frac{0.5}{10.875} = 230$ lbs. per square inch. The proper diameter of the stay-bolts, at the base of the thread, by formula (35), is equal to $.0069 \times 12 \times \sqrt{230} = 1.26$ inches, corresponding to the external diameter, 1½ inch.

IV. RESULTS OF EXPERIMENTAL TESTS OF THE RESISTANCE OF STAY-BOLTS AND FLAT STAYED-PLATES.

Screwed Stay-bolts.—Sir William Fairbairn,[1] in 1853, tested the strength of ¾-inch stay-bolts with enlarged ends screwed into ⅜-inch plates of copper and of iron, some of them being riveted or headed in addition. Figs. 260.

	Bolts.	Plates.	Breaking Weight.
1.	Copper	into copper, screwed and riveted	7.2 tons.
2.	Iron	into copper, do. do.	10.7 ,,
3.	Iron	into copper, screwed only	8.1 ,,
4.	Iron	into iron, screwed and riveted	12.8 ,,

In the first test, the bolt broke through the shank; the sectional area was .44 square inch, and the ultimate resistance was at the rate of $(7.2 \times \frac{1}{.44} =)$ 16.4 tons per square inch. In the second test, the rivet-head was broken off, and the bolt was drawn out of the plate, stripping the thread of the plate. In the third test, the thread of the plate was stripped by the bolt. In the fourth test, the bolt broke through the shank, the screw and the plate remaining uninjured. The resistance was at the rate of $12.8 \times \frac{1}{.44} = 29.1$ tons per square inch.



Figs. 260.—Flat Stayed-plates, ⅜ inch thick; Stay-bolts, ¾ inch.

Flat Stayed-plates.—The foregoing tests of holding power are confirmed by Sir William Fairbairn's experiments with flat stayed-plates tested by hydraulic pressure. He constructed two flat boxes, fig. 261, 22 inches square and 3 inches thick, having top and bottom plates of ½-inch copper and ⅜-inch iron respectively, inclosing a 2⅛-inch water space; stayed with $^{13}/_{16}$-inch iron stays, having enlarged ends screwed and riveted into the plates, to represent the conditions of the fire-box of a locomotive. The stays were placed at intervals of 5 inches in the first box, and of 4 inches in the second. They were tested, by hydraulic pressure, for bursting resistance.

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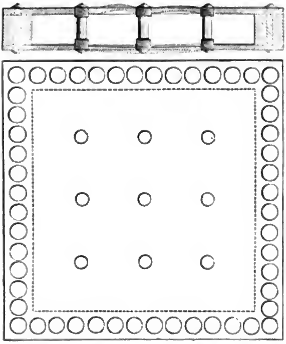
[1] See *Useful Information for Engineers* (1st series). London, 1856. Appendix, p. lxxiii.



Figs. 261.—Flat Stayed-plates, 5 inches pitch.—By Sir Wm. Fairbairn.

The first box bore a pressure which was gradually increased from 245 lbs. to 425 lbs. per square inch. The bulging or deflection was not noted until a pressure of 455 lbs. per square inch was attained, when the deflection amounted to .03 inch, or $^1/_{32}$ inch. At a pressure of 785 lbs. the deflection was .08 inch, or $^1/_{12}$ inch, and the box gave way at 815 lbs. by the pushing of the copper plate off the end of the central stay-bolt, and the stripping of the copper thread. The ultimate stress amounted to 9 tons on each stay-bolt.

The second box, under a pressure of 515 lbs., showed a deflection of .04 inch, or $^1/_{25}$ inch; at 965 lbs., and up to 1295 lbs., the deflection, as observed, stood without variation at .09 inch, or $^1/_{11}$ inch. At 595 lbs. the deflection amounted to .34 inch, or ⅓ inch; and at 1625 lbs. the box failed, by the pushing of the iron plate off the end of one of the middle stay-bolts, when the screw-thread in the plate was stripped. The ultimate stress on the stay-bolt was 11.6 tons.

It is deducible from the results of Sir Wm. Fairbairn's experiments that iron stay-bolts are 50 per cent stronger than copper bolts; that iron stay-bolts in iron plates are about 75 per cent stronger than copper bolts in copper plates; and that by riveting over the resistance is increased about 33 per cent, compared with bolts which are only screwed into the plate.

Also, that the fastening of a ¾-inch stay-bolt screwed into a ⅜-inch plate of the same metal as the bolt, and riveted, is stronger, when fairly tested, than the shank of the bolt; and that, generally, such a fastening for a bolt, of which the diameter is twice the thickness of the plate, of the same metal, is stronger than the shank. This deduction is corroborative of the argument, that the fastening and the shank would be equally resistant if the bolt be only screwed into the plate. This argument is, moreover, directly confirmed by the result of Sir William Fairbairn's experiment, in which a ¾-inch iron bolt stripped the thread of the screw from a ⅜-inch copper plate. Taking the diameter over the thread of the bolt at ⅞ inch, the area of the shorn surface was ($\frac{7}{8} \times 3\frac{1}{7} \times \frac{3}{8} =$) 1 square inch; the shearing resistance was 8.1 tons, or 50 per cent of the tensile strength of the bolt, which was proved, by the result of the first experiment, to be equal to 16.4 tons per square inch.

To compare the resistances of the stay-bolts when tested by themselves, and when tested in the chamber, it may be assumed that the hold of the bolt in the ½-inch copper plate was as much as in the ⅜-inch iron plate, since both plates failed, one in each box. The relative behaviours were then, as follows:—

Three-quarters inch Iron Stay-bolt screwed into a ⅜-inch Iron Plate, and riveted.

	Failure.	Final Stress per square inch.
		tons.
Tested separately	bolt broke in shank	12.8
In chambers, at 4 inches pitch	plate forced off	11.6
„ at 5 „ „	plate forced off	9.0

Here it is seen that the stay-bolts proved nearly as resistant,—within 10 per cent,—in the chamber, at the 4-inch pitch, as they did when tested separately; but that, at the 5-inch pitch, their resistance was less than three-fourths of that of the separate bolt: clear evidence of the advantage of a shorter pitch over a longer pitch, in bringing into action the shearing resistance of the bolt.

To apply the formula (28), for the strength of these two chambers: In the first chamber, with a 5-inch pitch, the bolts were, say, $^{13}/_{16}$ inch in effective diameter over the thread, and they were $(5 - ^{13}/_{16} =)$ 4.2 inches clear. The elastic resistance of the stayed-iron plates was equivalent to a pressure of $(5000 \times ⅜ \div 4.2 =)$ 446 lbs. per square inch. For the 4-inch pitch, the clear distance between the bolts was 3.2 inches, and the elastic resistance was equivalent to $(5000 \times ⅜ \div 3.2 =)$ 586 lbs. per square inch. Comparing the calculated strengths with the recorded data:—

	Bulging first noted at	Calculated Elastic Strength.	Stay-bolt Failed at.
	lbs.	lbs.	lbs. per sq. in.
5-inch pitch	455	446	815
4-inch pitch	515	586	1625

The calculated elastic strengths are, certainly, safely within the limits of ultimate strength. For the 4-inch pitch, the elastic strength is little more than one-third of the limiting strength; whilst, if the stay-bolt at the 5-inch pitch had been as effective as the others, the bursting pressure would have been greater in the ratio of 9 tons to 11.6 tons, already noted, or equal to $(815 \text{ lbs.} \times 11.6 \div 9 =)$ 1050 lbs. per square inch. The elastic deflections were by formula (25), for the 5-inch pitch $(4.2 \div 44 =)$ .10 inch; and for the 4-inch pitch $(3.2 \div 44 =)$ .073 inch, or $^{1}/_{14}$ inch. These values can scarcely be compared with the deflections recorded by Sir William Fairbairn, the observation of which was evidently wanting in precision.

Professor Thurston[1] records the results of experiments made in November, 1872, at Sandy Hook, on the resistance of stayed flat plates in a flat chamber 6 feet long, 4 feet high, and 4 inches thick, made of the Abbott Iron Company's "best flanged fire-box iron" plate, 5/16 inch thick, inclosing a 3⅜-inch water space. The stay-bolts, 1⅛ inches in diameter, were screwed into the back and the front plates at pitches of 8¾ inches by 9 3/16 inches, and slightly riveted over. The chamber, containing five-sixths of its capacity of water, was burst under steam, fires being built up on both sides. It exploded completely under a pressure of about 167 lbs. per square inch. "Both sides were stretched very considerably, assuming a dished form of 8 or 9 inches depth, and all of the stay-bolts drew out of the sheets without fracture, and without even stripping the thread of the external or the internal screw." On an area of (8¾ × 9 3/16 inches =) 80½ square inches per stay-bolt, the total resistance for one stay-bolt amounted to 6 tons.

The calculated elastic deflection, by formula (25), was $\left(\frac{9.2 - 1\frac{1}{8}}{44} =\right)$ .18 inch; and the elastic strength was, by formula (28), equivalent to (5000 × 5/16 ÷ 8.07 =) 193 lbs. per square inch. As a matter of fact, the chamber was exploded by a pressure of only 167 lbs. per square inch. So much the worse for the fact. The inferiority of the resistance displayed by this model was evidently due to the weakness of the fastening of the stay-bolts—to the strength of which, according to the reports, the riveting contributed very little—together with the wideness of the pitch of the stay-bolts. This experiment is instructive in showing how weak such an ill-proportioned structure may be.

Mr. Lavington E. Fletcher[2] records the results of tests made in August, 1876, at Portsmouth, on an experimental flat stayed-chamber, 8 feet 7 inches long, 4 feet 8 inches high, and 6¾ inches thick, as in figs. 262. The plates at the back were of Lowmoor iron, 7/16 inch thick; in the front, they were of Staffordshire quality, partly ½ inch thick, and partly 7/16 inch thick. The stay-bolts were 1¼ inches in diameter in the body, and fully 1⅜ inches over the thread, with 11 threads to the inch. They were pitched at 15½ inches horizontally and 15¾ inches vertically, and had a head or a nut at the front of the chamber, and were riveted over at the back. As the chamber was peculiarly constructed for a particular purpose, the action at those stay-bolts only which were placed under normal conditions will be noticed here.

At the first test, on attaining a pressure of 95 lbs., the back plate was forced over the end of No. 2 stay, stripping off the metal that had been riveted over, but leaving the threads on the bolt and in the plates uninjured.

For the second test, the stays 1, 2, 7, and 8 were replaced by bolts

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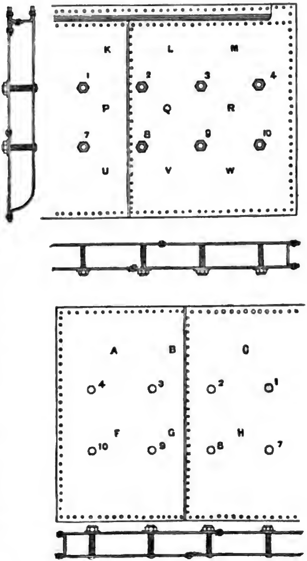
[1] *Journal of the Franklin Institute*, February, 1872; page 92.

[2] *Chief Engineer's Monthly Report to the Manchester Steam User's Association*, July-October, 1876, page 31.

1⅜ inch in diameter at the base of the thread, at both ends. At a pressure of 105 lbs. the back plate was forced over the end of No. 3 bolt; the threads were uninjured, but the metal that was riveted over was stripped off. For the third and fourth tests Nos. 3 and 9 stay-bolts were replaced by 1⅜-inch bolts, as before, with nuts at both ends. The back plate was forced over the end of No. 4 bolt, as in the second test, under a pressure of 155 lbs. For the fifth test, the two remaining stays, Nos. 4 and 10, were replaced by 1⅜-inch stays, as before, screwed and nutted. The front plate cracked at No. 3 hole, above and below, about 1½ inches each way, under a pressure of 150 lbs. For the sixth test, the leakage was stopped with india-rubber washers, and under 165 lbs. pressure the front plate burst at the lower line of rivets, through thirteen consecutive rivet-holes, which were countersunk.

The bulgings and sets noted when the pressure reached 60 lbs., 75 lbs., and 165 lbs. successively, are given in table No. 141. They "were gauged by the help of fine strings carried across the face of the chamber at fixed distances, the strings forming datum-lines from which the measurements were taken"—before pressure, while under pressure, and after pressure was withdrawn.



Figs. 262.—Experimental Flat Stayed-chamber, Portsmouth.

It appears from these results that the chamber burst, by slipping over stay-bolts, screwed and riveted only, at pressures of from 95 lbs. to 155 lbs. per square inch. When all the stays had nuts, the front plate cracked at No. 3 hole under a pressure of 150 lbs.; and ultimately it bore 165 lbs. without any failure at the stay-bolts. In testing the original marine boiler on which the chamber was modelled, the maximum amount of bulging in the plates of the boiler under pressures of from 60 lbs. to 65 lbs. per square inch, was 3/16 inch, in the plates corresponding to the front of the model chamber; and 1/16 inch full at the shell, represented by the back of the chamber. Of seven boilers tested in this way, only one showed any degree of set: 1/16 inch on the side of the uptake. These results harmonize with those deduced from the action of the model chamber. The working pressure of the boiler was 30 lbs. per square inch.

Table No. 141.—Bulging and Set of Flat Stayed-chamber, under Hydraulic Pressure. (Portsmouth.)

Back of Chamber. Stay-bolts 1, 2, 3, 4, 7, 8, 9, 10 Screwed through the Plate, and Riveted over at Ends.						Front of Chamber. Stay-bolts 1, 2, 3, 4, 7, 8, 9, 10 Screwed through the Plate and Nutted.					
Gauged at	Bulging.		Sets.			Gauged at	Bulging.		Sets.		
	At 60 lbs.	At 75 lbs.	At 60 lbs.	At 75 lbs.	At 165 lbs.		At 60 lbs.	At 75 lbs.	At 60 lbs.	At 75 lbs.	At 165 lbs.
	inch.	inch.	inch.	inch.	inches.		inch.	inch.	inch.	inch.	inches.
A	3/16	1/2	1/8 *f*	3/8 *b*	1 3/16	M	1/8	3/16	1/32 *b*	1/16 *f*	5/8
B	1/8	3/4	0	3/8 *f*	1 7/16	L	1/16	5/16	1/16	3/16	5/8
C	3/16	1/2	1/8	3/8 *f*	1 *f*	K	3/16	1/4	1/16	3/16	1/2 *b*
F	1/8	3/4	1/16 *b*	1/8	7/8 *b*	R	1/8	3/8	×	1/8 *f*	11/16 *f*
G	1/8	5/16	1/32 *f*	1/4	3/4 *f*	Q	3/16	5/16	×	1/4 *f*	5/8 *f*
H	1/8	3/8	1/16 *f*	5/16 *f*	11/16	P	1/8	5/16	×	1/8 *b*	3/8
						W	3/16	1/4	×	1/16 *f*	13/16 *b*
						V	1/8	3/16	×	1/8 *f*	1 1/8 *f*
						U	1/8	5/16	×	1/8 *f*	13/16 *b*

Note to Table.—*f* signifies *full*; *b* signifies *bare*.

The area supported by each stay amounted to (15½ × 15¾ =) 244.5 square inches; and the total pressures resisted by the stays were:—

	Lbs.	Tons.
In the 1st test.............	(244.5 × 95 =)	10.4 per stay-bolt.
„ 2d „	(244.5 × 105 =)	11.5 „ „
„ 4th „	(244.5 × 155 =)	16.9 „ „
„ 6th „	(244.5 × 165 =)	18.0 „ „

In the sixth trial, there was no failure at the stay-bolts. All the riveted ends of bolts, 1¼ inches in diameter at the base of the thread, supported a pressure of 17 tons, and no doubt they were capable of supporting 18 tons. The riveted bolts failed with from 10 to 17 tons.

A second chamber of the same dimensions as the one just described, with 7/16-inch front plates, and ½-inch back plates, was constructed and tested by hydraulic pressure at Portsmouth, in August, 1876. A screwed and riveted bolt was drawn out of the front plate, and the thread was stripped by a pressure of 120 lbs. per square inch. Two others were drawn through the back plate with 125 lbs. and 130 lbs. respectively. The fourth that failed was a riveted bolt, under 140 lbs. Finally, the chamber cracked at a rivet-joint, under a pressure of 140 lbs.

The elastic deflection of the Portsmouth chambers is, by formula (25), (15.75 − 1.25 ÷ 44) = .35 inch, an amount consistent with the average of the recorded observations for deflection, which are very various. The ultimate elastic pressure is, by formula (28), ($5000 \times 7/16 \div 15.5 =$) 141 lbs. per square inch. The chambers were burst under pressures of 195 lbs. and 140 lbs. respectively: and, in each instance, the failure took place at a rivet-joint at the edge of one of the plates.

Mr. Wm. Boyd[1] tested the resistance of flat stayed-plates of iron and of Siemens steel. Four boxes were constructed, 27 inches square and 9 inches deep, on angle-iron riveted frames, of which two were made with $\frac{1}{2}$-inch iron sides, and two with $7/16$-inch steel sides. The plates were connected by nine stay-bolts of iron and of steel respectively, at 9 inches pitch, screwed into both plates, in which the holes were drilled, and riveted over the plates in one iron box and one steel box, and nutted in the other iron box and the other steel box. The stay-bolts of the iron boxes were $1\frac{1}{2}$ inches in diameter over the thread, and $1\frac{3}{8}$ inches at the base of the thread; those of the steel-faced boxes were $1\frac{3}{8}$ inches and $1\frac{1}{4}$ inches over and under the thread. The diameter of the riveted head was $7/16$ inch larger than that of the bolt, and the thickness at the centre was $3/16$ inch. The nuts of the iron bolts were $1\frac{1}{8}$ inches deep; and those of the steel bolts 1 inch deep. The washers under the nuts were 3 inches in diameter, and $3/32$ inch thick.

The bulging was noted at several points on each side of each box, by measurement from a steel straight-edge attached to the framework, amongst which were the points A and B, the intersections of diagonals on one side of each box. The extent and increase of the bulging or deflection for each box at the points A and B were as follows:—

1. *Iron Box—Stay-bolts riveted over.*

Pressure in lbs. per square inch.

	65 lbs.	130,	195,	260,	325,	390,	422,	475 lbs.
	Deflection, in inches.							
At A,...	0,	.015,	.031,	.031,	.046,	.063,	.093,	.250 in.
At B,...	0,	0,	.015,	.015,	.045,	.093,	.124,	.406 ,,

2. *Steel Box—Stay-bolts riveted over.*

Deflection, in inches.

At A,...	0,	.031,	.031,	.046,	.093,	.390,	.406,	.406 in.
At B,...	0,	.031,	.031,	.046,	.093,	.281,	.408,	.531 ,,

3. *Iron Box—Stay-bolts nutted.*

Pressure in lbs. per square inch.

	65,	130,	195,	260,	325,	390,	455,	520,	585,	650,	715,	780 lbs.
	Deflection, in inches.											
At A,.....	—	—	—	—	.015,	.062,	.062,	.077,	.094,	.125,	.187,	.375 in.
At B,.....	—	—	—	—	—	—	.015,	.031,	.062,	.125,	.187,	.375 ,,

4. *Steel Box—Stay-bolts nutted.*

Deflection, in inches.

At A,.....	—	—	.031,	.031,	.031,	.093,	.218,	.265,	.407,	.531,	.656 in.	
At B,.....	—	—	—	—	.094,	.156,	.279,	.375,	.437,	.562,	.625 ,,	

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[1] "On Experiments relative to Steel Boilers," *Proceedings of the Institution of Mechanical Engineers*, 1878.

The ultimate failure of the boxes with riveted stays, by bursting, took place at the central stay-bolts, by the stripping of the edge or "lip" of the riveted head, which flew off as a ring; the threads of the screws were not damaged. The riveted boxes were burst by the failure of the angle-iron framing. Some of the nuts of the iron box were split. None of the stay-bolts were broken, nor were any of the plates cracked or fractured, nor were any of the washers bent outwards at the edge.

It appears that, under the actual conditions, the elastic resistance of the iron plates was greater, and was continued for a longer range of pressure, than that of the steel plates; whilst the deflection of the steel plates was greater than that of the iron plates. These differences are partly accounted for by the difference in thickness of the plates. For calculating the elastic strengths and the elastic deflections, by formulas (28) and (29), page 630; and (25), page 629, the following are the data:—

For No. 1 box......................$t = \frac{1}{2}$ inch, $d = (9 - 1\frac{1}{2} =)\ 7.5$ inches.
„ No. 2 do......................$t = \frac{7}{16}$ „ $d = (9 - 1\frac{3}{8} =)\ 7\frac{5}{8}$ „
„ No. 3 do......................$t = \frac{1}{2}$ „ $d = (9 - 3 \ \ =)\ 6$ „
„ No. 4 do......................$t = \frac{7}{16}$ „ $d = (9 - 3 \ \ =)\ 6$ „

For the 3d and 4th boxes, the pitch is here diminished by the diameter of the washers, 3 inches, to give the net span. The elastic pressures are:—

For No. 1 box..................$p = 5000 \times \frac{1}{2} \div 7.5 = 333$ lbs. per sq. in.
„ No. 2 do..................$p = 5700 \times \frac{7}{16} \div 7\frac{5}{8} = 327$ „ „ „
„ No. 3 do..................$p = 5000 \times \frac{1}{2} \div 6 = 417$ „ „ „
„ No. 4 do..................$p = 5700 \times \frac{7}{16} \div 6 = 416$ „ „ „

The elastic deflections are:—

For No. 1 box....................................$\delta = 7.5 \div 44 = .17$ inch.
„ No. 2 do....................................$\delta = 7.625 \div 44 = .173$ „
„ No. 3 do....................................$\delta = 6 \div 44 = .136$ „
„ No. 4 do....................................$\delta = 6 \div 44 = .136$ „

For the steel boxes, the calculated values correspond fairly with the yielding points, where the limits of elasticity are reached; for the iron boxes, they lie within the actual elastic range—considerably so for the nutted stay-bolts. The pressures at which deflection commenced, and set commenced, together with the bursting pressures, as stated by Mr. Boyd, are here subjoined for comparison:—

Box.	Deflection Begun.	Set Begun.	Elastic Pressure, calculated.	Bursting Pressure.
No. 1. Iron, riveted......	195 lbs.......	390 lbs.......	333 lbs.......	550 lbs.
„ 2. Steel, „	130 „	325 „	327 „	550 „
„ 3. Iron, nutted,......	390 „	585 „	417 „	1000 „
„ 4. Steel, „	260 „	390 „	416 „	900 „

Here, the calculated elastic pressures for the steel boxes are nearly the same as those at which the sets began; whilst, for the iron boxes, they are considerably less. In all instances, the elastic pressures, calculated, lie well within the actual bursting pressures; although these bursting pressures are not measures of the ultimate strengths of the stayed-plates. The ultimate strength of each box lay far beyond the calculated elastic limit.

The elastic deflections of the boxes,—at the elastic limits of pressure,—are here repeated, and the actual deflections, measured from the diagrams, are added, for comparison:—

Box.	Elastic Deflection, calculated.	Elastic Deflection, measured.
No. 1. Iron, riveted	.17 inch	.050 inch.
„ 2. Steel, „	.173 „	.094 „
„ 3. Iron, nutted	.136 „	.062 „ .015 „
„ 4. Steel, „	.136 „	.175 „

In each instance, except the last, the measured deflection was less than the calculated deflection. But the measured deflections are very various amongst themselves.

Messrs. David Greig and Max Eyth[1] record the results of tests of flat stayed surfaces, representing the sides of a locomotive fire-box. Three drums were constructed 35½ inches in diameter inside, with two diaphragms 2½ inches apart, one of which was ⅜ inch thick, and the other 9/16 inch thick. The stay-bolts were ⅞ inch in diameter, pitched 4¼ inches square, and were tapped right through both plates, having 14 threads per inch. Drums 1 and 2 were of steel and iron respectively, and in these the stay-bolts were riveted over at each end. In drum 3, which was of steel, check nuts were screwed on to the stay-bolts over the thinner diaphragm-plate. The edges of the diaphragms were flanged and riveted to the barrel or cylindrical part, with ¾-inch steel rivets in drilled holes, for the sake of strength. The drums were tested by hydraulic pressure, and gave way in each case by the breaking of a stay-bolt, in one of the outer rows, after the ⅜-inch diaphragm had acquired a permanent set. Messrs. Greig & Eyth consider that the bolts which broke were unfavourably placed, being in the neighbourhood of the comparatively yielding flanged corner of the plate. The area of plate supported by each bolt was 18.06 square inches. The diameter of each bolt at the base of the thread was barely 13/16 inch, giving .51 square inch of sectional area in the iron bolts, and .52 square inch in the steel bolts.

Drum	No. 2, Iron	No. 1, Steel	No. 3, Steel.
Breaking pressure per square inch	1230 lbs.	1628 lbs.	1445 lbs.
Corresponding stress on each stay-bolt	9.92 tons	13.16 tons	11.65 tons.
Do. do. per sq. in.	19.45 „	25.31 „	22.40 „
Ultimate tensile strength of stay-bolts	22.23 „	28.83 „	28.83 „

In the iron barrel, No. 2, the first signs of failure appeared with about 1150 lbs. pressure per square inch, when the edges of the stay-head over the ⅜-inch plate crumbled away. The plate was bulging out between the stays, and was beginning to leave, and partly to strip, the thread. Subsequently the 9/16-inch plate, at the other side, bulged freely, and the stay-bolt slipped gradually inwards through the ⅜-inch plate. In drum 1, the stay-bolt broke in the middle, between the two plates, after which the plates bulged freely, and leaked at the bolt-heads. In drum 3 also the stay-bolt broke in the middle.

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[1] See "Experiments referring to the use of Iron and Steel in High-pressure Boilers," in the *Proceedings of the Institution of Mechanical Engineers*, 1879.

The calculated elastic limits of pressure, by formulas (28) and (29), were for,—

No. 2. ⅜-inch plate .... 634.1 lbs. per square inch.
No. 1. Do. .... 555.6 „ „
No. 3. Do. .... 634.1 „ „

These quantities are consistent with the observed breaking stresses; they are from one-half to one-third of these.

The records of bulging deflection of the diaphragms are, so far as they can be analysed, less than the deflections calculated by formulas already given.

Mr. Phillips tested two experimental flat boxes at Plymouth Dockyard, representing marine practice. They were constructed respectively of 7/16-inch and of ½-inch iron plates, stayed at intervals of 15¾ inches by 15¼ inches, with 1⅜-inch screwed stay-bolts riveted over at the ends, giving a superficies of 240 square inches for each bolt. Tested by hydrostatic pressure, the plates were bulged between the stay-bolts, and were finally pushed off, or drawn away from the bolts, under the following pressures:—

PLATES.	Sectional Area of Stay-bolts.		Bursting Pressure per Square Inch of Surface.	Total Pressure for each Stay-bolt.
	In Body.	At Thread.		
inch.	square inch.	square inch.	lbs.	tons.
7/16	1.48	(say) 1.2	105	11.25
½	1.48	(say) 1.2	140	14.73

When nuts were applied to the ends of the stay-bolts through the 7/16-inch plate, they bore a pressure amounting to 165 lbs. per square inch, on the plate; when the box gave way at a riveted joint.

The maximum resistance of these plates has evidently not been developed. The elastic strength of the 7/16-inch plate, by formula (28), is $(5000 \times 7/16 \div 14 =)$ 157 lbs. pressure per square inch.

Conclusion.—The calculated elastic strength, in all cases except the Portsmouth and Plymouth trials, lies well within the range of the actual bursting resistance. In these trials the failures arose, not from the weakness of the plates, but from that of the rivet-joints, or of the stay-fastenings, which were loosened and weakened by the bulgings of the plate round the stay-bolts. The greater degree of resistance of the stay-bolts of Sir William Fairbairn's boxes was clearly derived from the comparative nearness of the stay-bolts in those boxes, and the more limited range of the distortion to which the stay-fastenings were subject. Even between these two boxes, the advantage of the closer 4-inch pitch over the wider 5-inch pitch, was perceptible; for the more closely set stay-bolts made the greater resistance to bulging stress. Whilst the 5-inch stay-bolts resisted rupture with a force of 9 tons, the 4-inch stay-bolts resisted with a force of 11½ tons.

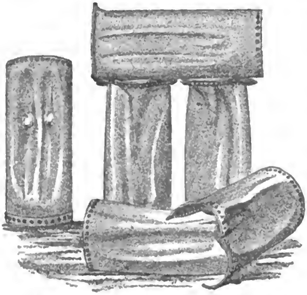
Upon the whole, the values for elastic deflection and elastic strength of flat-stayed surfaces given by means of the formulas may be accepted as approximately correct values, on the assumption that the stay-bolt fastenings are sufficiently strengthened to resist the weakening influence of distortion other than that which arises from direct stress fairly applied. The nuts of the stay-bolts employed in the Portsmouth boxes proved to be effectual for securing that object.

CHAP. II.—ELEMENTAL STRENGTH OF STEAM BOILERS—FLUE-TUBES, SEGMENTAL CROWNS.

I. STRENGTH OF FURNACE FLUE-TUBES.

The resistance of plain cylindrical flue-tubes is usually supposed to be derived partly from the longitudinal tension to which they are supposed to be normally subject, as between the two fixed plates between which they are suspended, when deflection or alteration of form commences; and partly from the direct resistance to compression and collapse, in virtue of the cylindrical form of the flue. On this supposition, when the tube springs or incipiently collapses, the collapsing part is subjected to tensional stress, by the fact of its deflection from a straight line between the extremities. But the actual conditions are not according to this supposition. The furnace is, by its situation, of a temperature much higher than that of the shell, and it endeavours to expand longitudinally to a greater extent than the shell does; but, fastened to the ends, it is hindered from expanding to its natural limit, and is thus subjected to longitudinal compression. Accordingly, when it springs or incipiently collapses locally, the elongation of the part affected, in the first place, simply gives relief from compressive stress, without deriving any end support; and only when it is sufficiently elongated by deflection is it assisted by tensional stress. That furnace-tubes are thus unsupported from the ends, is demonstrated by the uniform flatness of the figure of thoroughly collapsed flue-tubes of considerable length, and the sharp slopes with which each end of the tube joins the end-plate. The usual rate of slope is about 1 radially to 2 or 2½ horizontally; or, otherwise, the axial length of the funnel-shaped junction at each end of a long collapsed flue is equal to the diameter, or 1½ times the diameter, of the flue. From this it would follow that the support from the two ends together does not, under any circumstances, practically extend over a length of tube greater than twice or three times the diameter.

An instructive case of partial collapse of short furnace-tubes resulted from the overheating of a number of flues made of Farnley iron, being the furnace-tubes of some marine boilers in which, when at sea, the water-level was allowed to become too low. The flues became red hot, and partially collapsed. A few of them are represented by figs. 263, of which the lengths are from two to three times the diameter. The circular form of the ends was maintained by the fastenings; but trough-like depressions were formed by the relative over-pressure of the steam—in one instance of a depth nearly half the diameter. The intervening ridges are portions of the original formation, and evidently served as struts in compression, taking the stress of the depressions in tension. These instances are confirmatory of the want of longitudinal resistance to collapse, even in comparatively short lengths of, say, 6 feet or 9 feet. Hence the propriety of limiting the segments of flue-tubes joined together with flange-seams, to lengths of about 3 feet, the better to sustain the tube in shape, in case of incipient collapse.


Figs. 263.—Collapsed Furnace-tubes of Farnley Iron.

It follows that the hydrostatic test is not conclusive on the power of resistance of furnace-tubes to collapse under steam-pressure; seeing that the shell and the tube, under the hydrostatic test, are of uniform temperature; and are not subjected to the unequal expansion and cross stress incidental to the steam test.

Reverting to the supposition that the resistance of cylindrical furnace-tubes is derived partly from longitudinal tension between the ends, the possible aid thus derivable may be estimated. Conceive a longitudinal strip of the flue, 1 inch wide, to be isolated and suspended between the ends, and subjected to uniform pressure. The line of argument adopted, for calculating the strength of flat end-plates, page 620, is applicable here. The ultimate elastic deflection is, modifying formula (5), page 622,

$$\delta = \frac{l}{3.67} \quad \ldots\ldots\ldots\ldots \quad (1)$$

in which δ = the deflection in inches at the mid-length, and l = the length of the strip, or—in this case—of the flue, in feet. Conceiving the strip to take approximately the form of a segment of a circle, the diameter D of this circle = 11 l, by formula (7), page 622, and the elastic stress in the strip, strained to the elastic limit, is

$$p = \frac{4480\,t\,s}{11\,l \times 12} = \frac{34\,t\,s}{l} \quad \ldots\ldots\ldots\ldots \quad (2)$$

p = the pressure of stress in pounds per square inch.
t = the thickness of the plate in inches.
s = the tensile stress in the plate, in tons per square inch, at the elastic limit.
l = the length of the strip in feet.

Here, it may be noted, that the coefficient, 4480, in the formula (2), is one-half of that of the similar formula (10) for end-plates, page 622; as the bursting resistance of a cylinder transversely is only one-half of that of a sphere. Substituting the values of s, 12 tons and 14 tons for iron and steel respectively, the formula becomes $p = \frac{34t \times 12}{l}$ for iron; and reducing, it becomes, for iron and for steel:—

$$\text{For iron,} \ldots\ldots \quad p = \frac{400\,t}{l} \ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

$$\text{For steel,} \ldots\ldots \quad p = \frac{470\,t}{l} \ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

The resistance to collapse is directly as the square of the thickness of the plate, and inversely as the square of the diameter. The first ratio is obviously true; and, for the second, whilst the leverage of the pressure to double up the plate increases as the diameter of the flue, the pressure also increases as the diameter; and the stress is in the combined ratio of leverage and pressure, or as the square of the diameter.

To form an idea of the assistance possibly derivable from longitudinal tension in resisting collapse, take an instance of an iron flue-tube 25 feet long, 36 inches in diameter, of ⅜-inch plate. By formula (1), the ultimate elastic deflection of a flat strip, under the given circumstances, is $\delta = \frac{25}{3.67} = 6.8$ inches; and the pressure, formula (3), by which such deflection would be produced is $\frac{400 \times \frac{3}{8}}{25} = 6$ lbs. per square inch. A deflection of 6.8 inches is inadmissible for useful resistance; and if even 1 inch of deflection, within working limits, be allowed, the corresponding pressure would not exceed 1 lb. per square inch.

Take, again, a segment of a flue of ⅜-inch plate, made with flanged joints, 3 feet in length. The deflection of a 1-inch strip within the elastic limit would be $\frac{3}{3.67} = .82$ inch; and the acting pressure would be $\frac{400 \times \frac{3}{8}}{3} =$ 50 lbs. per square inch.

But, unfortunately, the co-operation of longitudinal resistance can only be brought into action by such a deviation from the cylindrical form as to seriously impair or even destroy the direct compressive resistance to collapse. Therefore, in the case of considerable lengths of flue-tube, the longitudinal resistance, except within very narrow limits, would not be available as a security against collapse. But, let us assume an arbitrary limit of deflection, or deviation from the normal form, of 1 inch, for the exertion of longitudinal tensional resistance. The following are the relative pressures per square inch that would be supported and resisted under that condition, for various lengths of plain iron flue-tube:—

Length of Tube.	Deflection.	Pressure Per Square Inch Resisted by Longitudinal Tension, for Several Thicknesses of Plate.		
		⅜ inch.	7/16 inch.	½ inch.
feet.	inches.	lbs.	lbs.	lbs.
3	.82	50	58	67
6	1.00	15	17.5	20
9	1.00	7	8	9
12	1.00	4	4.75	5
15	1.00	2.5	3	3.5
18	1.00	1.7	2	2.2
21	1.00	1.2	1.5	1.6
24	1.00	1	1.2	1.3

From this statement it appears that the longitudinal resistance to collapse of plain flue-tubes, though important, on the given supposition, for short lengths of from 3 feet to 6 feet or 7 feet, is insignificant for greater lengths; and that, therefore, all the resistance to collapse of unfortified lengths of plain flue-tubes more than 6 feet or 7 feet long is supplied substantially by the transverse compressive strength and stiffness of the tube. The current practice of boiler manufacturers and steam-users is corroborative of the foregoing conclusions on the resistance of furnace-tubes. From a list of 2000 plain unfortified iron furnace-tubes of Lancashire and Cornish steam boilers, recently prepared by Mr. Michael Longridge, a random selection has been made by the author of eighty-four tubes, of ⅜-inch, 7/16-inch, and ½-inch plates, of lengths varying from 7 feet to 32 feet, together with the working steam-pressures which they had supported for years, without a failure. The several pressures for each thickness of plates do not seem to have any relation to the lengths of the tubes. But in the following table, No. 142, of particulars, there is a sensible relation between the diameters, lengths, and pressures, taken together, and the thickness of plates; as is clearly shown by the comparative products of these three elements, thus:—

⅜-inch plates……………….29.5 × 19.3 × 41 = 23,343
7/16 " ……………….33 × 20 × 45.5 = 30,030
½ " ……………….33 × 24.6 × 46 = 37,342

Table No. 142.—PLAIN IRON FURNACE-TUBES OF LANCASHIRE AND CORNISH STEAM BOILERS, IN WORKING CONDITION.

Thickness of Plates.	Number of Tubes.	Diameter.	Average Diameter.	Length.	Average Length.	Working Pressure per Square Inch.	
						Pressures.	Average.
inch.	tubes.	inches.	inches.	feet.	feet.	lbs.	lbs.
⅜	43	21 to 36	29.5	10.6 to 30	19.3	30 to 60	41
7/16	34	27 to 44	33	7 to 32	20	18 to 65	45.5
½	7	30 to 42	33	11 to 32	24.6	30 to 66	46

These products are approximately as 3, 4, 5, or approximately as the squares of the thicknesses of plates; and they seem to indicate that, taking comprehensive averages, the minds of designers were influenced in some degree by the element of length of tube, in relation to thickness of plates.

The following instances (table No. 143) of collapsed flues of Cornish and of Lancashire boilers have been collected by the author from Mr. Lavington E. Fletcher's reports.[1] They comprise flue-tubes of from 32 inches to 48 inches in diameter, ⅜ inch and 7/16 inch in thickness, and from 18 feet to 40 feet in length. The working pressure of steam at or about the time when each collapse took place is given in column 4, and the compressive stress on the metal per square inch of section is given in the last column. This stress is not generally so much as 1 ton per square inch. In some instances, no doubt, the flue-tubes had been sensibly worn; but, in most instances, they had been in good order.

Table No. 143.—Collapsed Flue-tubes of Cornish and Lancashire Boilers.

Unfortified Plain Tubes.

Length of Boilers. 1	Inside Diameter of Flue-tube. 2	Thickness of Flue-tube. 3	Working Pressure of Steam, in Lbs. per Square Inch. 4	Transverse Compressive Stress per Square Inch of Longitudinal Section of Tube. 5
feet.	inches.	inch.	lbs.	tons.
18	32	⅜	60	1.16
24	34	"	68	1.40
25	35	"	64	1.35
26	35	"	60	1.28
43	36	"	50	.82
28	36	"	45	.74
33¼	39	"	45	.80
25¼	41	"	40	.74
28	42	"	45	.86
30¼	42	"	45	.86
32	44	"	40	.80
34	48	"	40	.87
17	34	7/16	55	.87
32	43	"	57	1.12
40	45	"	45	.92
32	47	"	50	1.07
30	48	"	43	.94
33	48	"	42	.92
32	48	"	50	1.10
28	48	"	50	1.10

It is plain from this table that the mere length of flue-tube did not influence the quantity of the resistance, which was simply the direct resistance to collapse. Plotting the pressures on ⅜-inch plate flues, column 4, as ordinates to a base-line representing the diameters, column 2, the medium curve traced through the plottings shows that the resistance to collapse varied inversely as the 1.75 power of the diameter: that is to say, not in so great a ratio as the square of the diameter. It shows that—

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[1] *Monthly Reports to the Manchester Users' Association*, 1862–69.

For diameters of................	30,	35,	40,	45,	50 inches, ⅜-inch plate,
The collapsing pressures were	76,	58,	45,	37,	30 lbs. per square inch.

The collapsing pressures for plain iron furnace flue-tubes ⅜ inch thick, are represented by the expression—

$$\frac{29{,}000}{d^{1.75}} \text{ lbs. per square inch} \ldots\ldots\ldots\ldots \quad (a)$$

Plotting likewise the results for flues made of 7/16-inch plate, though limited in range, they show that—

For diameters of................	40,	45,	50 inches, 7/16-inch plate,
The collapsing pressures were	60,	49,	42 lbs. per square inch,
Against the pressures...........	45,	37,	30 lbs. per square inch for ⅜-inch plate.

These values of actual collapsing pressures are considerably in excess of the working pressures in current practice, given in table No. 142, preceding. Thirty-inch tubes, for example, ⅜ inch thick, are worked with steam of 41 lbs. pressure per square inch; and they withstood 76 lbs. pressure before collapsing.

The collapsing pressures are relatively as the squares of the thicknesses of plate; and the numerator of the expression (a), is convertible into the value ($200{,}000\,t^2$), giving for the general equation of the resistance of plain iron flue-tubes to collapse,—

Collapsing Pressure of Plain Iron Flue-tubes of Cornish and Lancashire Steam Boilers.

$$p = \frac{200{,}000\,t^2}{d^{1.75}} \ldots\ldots\ldots\ldots \quad (5)$$

p = collapsing pressure, in pounds per square inch.
t = thickness of the plates of the furnace-tube, in inches.
d = internal diameter of the furnace-tube, in inches.

For shorter lengths, it has already been shown, page 646, to what extent the longitudinal tensile resistance may be effective in augmenting the resistance to collapse. It would follow that flues efficiently fortified by flange-joints or hoops at intervals of 3 feet may be enabled to resist from 50 lbs. to 60 lbs. or 70 lbs. pressure per square inch more than plain tubes, according to the thickness of the plates.

II. STRENGTH OF SMALL TUBES.

The resistance of solid drawn tubes of small diameters—from 3¼ inches to 1¾ inches in diameter, and from .134 inch to .109 inch in thickness—has been tested experimentally by Messrs. J. Russell & Sons.[1] The results are averaged and summarized as in the table No. 144.

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[1] See a notice of these experiments in *A Manual of Rules, Tables, and Data for Mechanical Engineers*, page 693.

Table No. 144.—EXPERIMENTAL RESISTANCE OF SOLID-DRAWN TUBES TO BURSTING PRESSURE AND COLLAPSING PRESSURE.

(Deduced from Messrs. Russell's data.)

WROUGHT-IRON TUBES.

External Diameter.	Thickness.		Internal Diameter.	Bursting Pressure.		Collapsing Pressure.		Difference of Bursting and Collapsing Pressures.
				Per square inch of Surface.	Per square inch of Section of Metal.	Per square inch of Surface.	Per square inch of Section of Metal.	
inches.	B.W.G.	inch.	inches.	lbs.	tons.	lbs.	tons.	tons.
3¼	10	.134	2.982	4800	23.84	3300	17.86	5.98
3⅛	10	.134	2.857	4500	21.42	3150	16.40	5.02
3	11	.120	2.760	4500	23.10	3500	19.53	3.57
2¾	11	.120	2.510	5200	24.28	3500	17.89	6.39
2½	11	.120	2.260	5000	21.02	3600	16.74	4.28
2¼	11	.120	2.010	5900	22.06	4500	18.82	3.24
2	12	.109	1.782	5900	21.53	4900	20.07	1.46
1¾	12	.109	1.532	5600	17.57	4000	14.33	3.24
Averages, omitting the last tube				—	22.40	—	18.20	—
HOMOGENEOUS METAL TUBES.								
3	13	.095	2.810	3600	23.77	3150	22.20	—
2¼	13	.095	2.060	7600	36.78	4600	24.32	—
2	15	.072	1.856	4000	23.02	3500	21.70	—
1⅝	17	.058	1.409	4600	24.94	4000	25.02	—
BESSEMER STEEL TUBES.								
1¾	13	.095	1.56	7800	28.92	4600	18.91	—
HYDRAULIC TUBES.				proved to				
2½	—	5/16	1⅞	11,000	14.73	—	—	—
1⅝	—	5/16	1	6,000	4.29	—	—	—
1⅛	—	¼	⅝	4,000	2.23	—	—	—
⅞	—	5/16	¼	12,000	2.15	—	—	—

III. STRENGTH OF SEGMENTAL CROWNS OF FURNACES.

The spherical-segmental upper parts or crowns of certain kinds of vertical boilers may be calculated for resisting force on the same principle as segmental ends have been treated, page 627. The action of the force is reversed—being compressive instead of tensile, and tensile instead of compressive; and, similarly, the resistance of the spherical segment to external pressure, acting compressively, is twice as much as that of a cylindrical body, even if it be thoroughly stayed to prevent failure by collapse.

The formulas (19) to (22), pages 627, 628, for the pressure, stress in, and elastic strength of segmental or spherical ends of cylinders, in resisting internal pressure, are applicable in the case of compressive resistance to external pressure, as follows:—

Relative External Pressure and Stress in the Segmental Crown.

$$p = \frac{8960\,ts}{\frac{r^2}{v} + v} \quad \ldots\ldots\ldots\ldots \quad (6)$$

Elastic Strength of Segmental or Spherical Crowns of Cylinders, of uniform thickness, for resisting uniform external pressure.

$$\text{Wrought iron,}\quad p = \frac{108{,}000\, t}{\frac{r^2}{v} + v} \quad \ldots\ldots \quad (7)$$

$$\text{Steel,}\ldots\ldots\quad p = \frac{125{,}000\, t}{\frac{r^2}{v} + v} \quad \ldots\ldots \quad (8)$$

$$\text{Cast iron,}\ldots\ldots\quad p = \frac{45{,}000\, t}{\frac{r^2}{v} + v} \quad \ldots\ldots \quad (9)$$

t = thickness of plate, in inches.
r = radius of circular junction, or of the cylinder, when not otherwise described, in inches.
v = versed sine, or rise of segment, in inches.
p = external pressure, in pounds per square inch.

In order that the results of these formulas may be implicitly accepted, it is indispensable that the spherical surface be perfectly formed, and the junction with the cylindrical portion truly circular; seeing that, whilst the tendency of internal pressure is to perfect an originally imperfect spherical form, external pressure tends to aggravate any deviation from a truly spherical surface. At the same time, it is much in favour of a spherical end or crown subjected to compressive pressure, that it possesses great power of resistance to buckling, and that the tendency of the strain is to render the circumferential junction, if originally imperfect in form, more nearly truly circular.

Again, the proportional rise or versed sine of the segment of equal elastic strength, supposing the plates of the crown and the cylinder of uniform thickness, may be settled by the formula (23) already used, page 628. Thus:—

Versed Sine of Segmental Crown of equal strength.

$$v = .27\, r \ldots\ldots \quad (10)$$

That is to say, the rise at the middle of the spherical segment or crown, of equal strength, having equal thickness of plates, is, say, one-fourth of the radius of the cylinder, or one-eighth of the diameter.

When the spherical segment is a semi-sphere, in which the rise is equal to the radius of the end, the plates, for an end of equal strength, need only be of half the thickness of the plates of the cylindrical part—supposing every part to be perfectly formed and constructed. But it is not recommended that the plates be of only half the thickness.

The compressive stress in the segment due to a given external pressure, and to given dimensions, is found by means of the formula (24), page 629, already stated for tensile stress:—

Compressive Stress in a Segmental Crown for a given pressure.

$$s = \frac{p\left(\frac{r^2}{v} + v\right)}{8960\,t} \quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (11)$$

Example.—The inverted dished plate *b*, in Mr. Nicholls' experimental cylinder, fig. 259, page 625, sustained a maximum pressure of 380 lbs. per square inch. What was the compressive stress to which the metal was subjected? Here $p = 380$ lbs., $t = \frac{3}{8}$ inch, $r = 15$ inches, and $v = 3\frac{1}{2}$ inches. Then,—

$$s = \frac{380\left(\frac{15^2}{3.5} + 3.5\right)}{8960\,t} = \frac{380(64.3 + 3.5)}{3360} = 7.67 \text{ tons per square inch.}$$

This high stress—7.67 tons per square inch—is far in excess of any resistance that was offered by plain cylindrical flue-tubes, as recorded in page 647, and it is confirmatory of the statement made that a segmental plate in such a situation possessed great power of resistance to buckling and consequent failure.

CHAP. III.—ELEMENTAL STRENGTH OF STEAM BOILERS—RIVETED JOINTS.

I. STRENGTH OF RIVETED JOINTS IN IRON PLATES.

It is well known that Sir William Fairbairn, in 1838, deduced from experiments with ¼-inch riveted plate-joints, that their strengths were relatively as 100, 70, and 56, for the whole plate, the double-riveted lap-joint, and the single-riveted lap-joint.

At Woolwich Dockyard, Mr. W. Bertram tested various iron plate-joints. The results were reported, and they were investigated by the author, in 1859.[1] Staffordshire plates of good quality were selected for the trials, of three thicknesses, ½ inch, 7/16 inch, and ⅜ inch; and made up into seven varieties of rivet-joints, 4 inches broad, 24 inches long, made with ¾-inch rivets at 2 inches of pitch. The double-riveted specimens were chain-riveted. Three specimens of each variety of joint were tested, and the results were averaged. The fractures took place, in nearly all cases, in one of the plates. The normal tensile strength of the whole plates was nearly uniform, and averaged 20 tons per square inch.

Mr. J. G. Wright[2] gives the strength of two specimens of single-riveted square lap-joints, and two of diagonal joints, at an angle of 45°, which were tested by Mr. Kirkaldy. They were made from ⅜-inch Staffordshire plates, 12 inches wide, with 2¼ inches of lap, 13/16-inch rivets at 2 inches of pitch.

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[1] See *Railway Locomotives*, 1860, by D. K. Clark; page 2*; also, a *Manual of Rules, Tables, and Data for Mechanical Engineers*, by the same; page 634.

[2] See discussion upon Mr. W. R. Browne's paper "On the Strength and Proportions of Riveted Joints," in the *Proceedings of the Institution of Mechanical Engineers*, 1872.

The diagonal joint was made with eight rivets of the same size and pitch. The ultimate tensile strength of the solid plate was 19.69 tons per square inch with the fibre, and 16.80 tons across. The section of the whole plate was (12 × .38 =) 4.56 square inches, and the total ultimate strength in the direction of the fibre was (4.56 × 19.69 =) 89.8 tons.

	Ultimate Tensile Strength.	
Entire plate	89.8 tons.	
Square joint	43.0 "	or 48 per cent.
Diagonal joint	58.0 "	or 64 "

	Square Joint.	Diagonal Joint.
Net sectional area	59.4 per cent.	91.7 per cent of entire plate.
Net tensile strength	48.0 "	64.0 " "
Do. per square inch of net section, in parts of that of the whole plate	81.2 "	70.5 " "

The diagonal joint was one-third stronger than the square joint; although, per square inch of net section, it opposed less resistance, because its resistance, which was necessarily exerted in an oblique direction, was a resultant compound of shearing resistance with the lengthwise resistance of the plates.

The net sectional area of the square joint was 2.71 square inches; and the shearing section of the rivets, 3.11 square inches, or 115 per cent of the net section.

Mr. Brunel made experiments on double-riveted double-welted plate-joints, specially suitable for girder-work, of which the author published an analysis in 1858–59.[1] The specimens were of best Staffordshire plates, ½ inch thick, 20 inches wide, butt-jointed, with a covering or fishing plate, or welt, on each side, 10 inches long, put together with punched holes and rivets, for chain riveting or square riveting, and zigzag riveting.

Five whole plates, from 12 to 16 inches wide, of the same quality as the specimens, were broken by a force of from 19.4 to 22 tons per square inch: average strength, 20.6 tons per square inch.

Mr. Longridge reported the results of tests for the strength of riveted joints in iron boiler-plates, made for him by Mr. Kirkaldy.[2] The plates were manufactured by the Snedshill Bar and Bolt Iron Company; and the rivet-iron and the rivets of scrap-iron, by Messrs. W. & J. Galloway & Sons, who also constructed the test-specimens. The specimens were made several inches in width, of ½-inch and 7/16-inch plates,—lap-jointed and with double-welted joints, single-riveted and double-riveted. Two specimens of each variety were tested, and the mean strength for each pair is reported.

The gain of strength due to annealing the plates of the single-riveted lap-joints was 9 per cent when the rivets were shorn, and 17½ per cent when the plate broke. For double-riveted lap, and single-riveted double-welt, the gains were 6½ per cent.

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[1] *Recent Practice in the Locomotive Engine*, 1858–59; also *Railway Locomotives*, 1860; page 2*.

[2] *The Boiler Insurance and Steam Power Company:* Chief Engineer's Report, 1877.

II. DEDUCTIONS FROM THE EVIDENCE ON IRON-PLATE RIVETED JOINTS.

The evidence supplied by the experiments of Bertram, Wright, Brunel, and Longridge, just noticed, is exhaustively analysed in a work now in course of preparation by the author on the strength of materials. Meantime, the deductions from the evidence are here given in summary. The numerous variations of dimensions and proportions render it somewhat difficult to draw useful deductions. But, to facilitate this work, a selection from amongst the best examples of their kinds is drafted into table No. 145. To these are added results of some experiments by Messrs. Greig and Eyth. (See foot-note, page 656.) The double-riveted joints are zigzag, except Nos. 12, 13, and 14, which are chain-riveted.

For the sake of simplicity of treatment, the original diameters of the rivets are employed in calculations of shearing area, and of the interspaces between rivets.

Single-riveted Lap-joints in Iron Plates. Table No. 145.—There are four instances of ⅜-inch plate-joints, Nos. 3, 4, 10, and 11. Of these, Nos. 3 and 11 have nearly equal breaking strengths, 56 per cent and 57.6 per cent. They have both ¾-inch rivets, but No. 11 has only 1⅞ inches of pitch against 2 inches in No. 3. The slightly greater strength of No. 11, notwithstanding the shorter pitch, may be due to the drilling of the holes, as against the punching in No. 3. No. 10, having ⅝-inch rivets and 1⅝ inches of pitch—making a slightly greater net plate-section than No. 11—has only 50.6 per cent breaking strength, against 57.6 per cent. The inferiority appears to be associated with the smaller size of rivet and the shorter pitch; and this surmise with respect to No. 10 is strengthened by the parallel case of No. 9, having 9/16-inch rivets and 1½ inches of pitch, with 62.5 per cent net plate-section. It has only the same breaking strength, 50.6 per cent, as No. 10 has, even though it has, in its favour, less thickness of plate. On the other extreme, No. 4, having 13/16-inch rivets, also has a relatively low breaking strength, 48 per cent. This is the result of an isolated experiment, the conditions of which are not recorded.

Of the ½-inch plate-joints, Nos. 1, 5, and 6, table No. 145, the respective breaking strengths are 40, 39.4, and 39 per cents. These are, consistently, almost identical, seeing that the proportions of rivet, pitch, and net plate-section, are very nearly the same: ¾-inch rivets at 2 inches of pitch, with 62.5 per cent net plate-section; and 13/16-inch rivets at 2⅛ inches pitch, with 61.8 per cent net plate-section. The larger diameter and pitch of rivets, namely, 13/16 inch and 2⅛ inches, appear to be the more suitable for ½-inch plates, as, without materially reducing the proportion of net plate-section, they provide a greater proportion of shearing section,—48.8 per cent of the whole plate-section, against 44.2 per cent for the ¾-inch rivets.

Of the 7/16-inch plate-joints, Nos. 2, 7, and 8, table No. 145, the respective

Table No. 145.—SUMMARY RESULTS OF CHARACTERISTIC TESTS OF RIVETED JOINTS OF IRON PLATES.

(Experiments of Bertram, Longridge, Greig and Eyth, &c.)

JOINT, AND REFERENCE	Punched or Drilled.	Nominal Thickness of Plate.	RIVETS.		Net Section of Plate, in parts of that of whole Plate.	SHEARING SECTION.		Fracture in Plate or in Rivets.	Breaking Strength, in parts of that of whole Plate.	Breaking Strength per Square Inch in parts of that of whole Plate per Square Inch.	
			Diameter.	Pitch.		In parts of Net Section of Plate.	In parts of whole Plate.			Of Net Plate-section.	Of Shearing Section.
1	2	3	4	5	6	7	8	9	10	11	12
SINGLE-RIVETED LAP-JOINTS.		inch.	inch.	ins.	per cent.	per cent.	per cent.		per cent.	per cent.	per cent.
1. Bertram, No. 16......	punched	½	¾	2	62.5	70.7	44.2	—	40	66.2	90.4
2. Do. No. 17......	punched	7/16	¾	2	62.5	80.8	50.5	—	52	85.7	103.0
3. Do. No. 18......	punched	⅜	¾	2	62.5	94.2	58.9	—	56	92.7	95.2
4. Wright	punched	⅜	13/16	2	59.4	115.0	68.3	—	48	81.2	70.6
5. Longridge, No. 2....	punched	½	13/16	2⅛	61.8	79.0	48.8	rivets, plate	39.4	63.8	80.7
6. Do. No. 4....	drilled	½	13/16	2⅛	61.8	79.0	48.8	rivets	39.0	63.1	79.9
7. Do. No. 6....	punched	7/16	¾ bare	2	62.5	80.6	50.4	rivets, plate	40.1	64.0	79.4
8. Do. No. 8....	drilled	7/16	¾ bare	2	62.5	80.6	50.4	rivets	41.6	66.6	82.6
9. Do. No. 15..	punched	¼	9/16	1½	62.5	106.0	66.2	rivets, plate	50.6	80.9	76.3
10. Greig & Eyth, Nos. 1 & 7	drilled	⅜	⅝	1⅝	61.5	81.8	50.3	rivets, plate	50.6	82.4	100.7
11. Do. No. 12	drilled	⅜	¾	1⅞	60.0	104.7	62.8	plate, rivets	57.6	96.0	91.7
DOUBLE-RIVETED LAP-JOINTS.											
12. Bertram, No. 19 (chain)	punched	½	¾	2	62.5	141.4	88.4	—	59	94.4	66.8
13. Do. No. 20 ,,	punched	7/16	¾	2	62.5	161.6	101.0	—	70	112.0	69.3
14. Do. No. 21 ,,	punched	⅜	¾	2	62.5	188.4	117.7	—	72	115.2	61.2
15. Longridge, No. 16...	punched	½	15/16	3	68.8	133.9	91.1	plate	60.3	87.7	65.5
16. Do. No. 17...	drilled	½	⅞	3	70.8	113.2	80.1	plate	64.9	91.7	81.0
17. Do. No. 21...	punched	7/16	¾ full	2½	70.0	115.4	80.8	plate	66.1	94.4	81.8
18. Greig & Eyth, No. 14	drilled	⅜	⅝	2½	75.0	87.2	65.4	plate zigzag, rivets	64.6	86.1	98.7
SINGLE-RIVETED SINGLE-WELT.											
19. Greig & Eyth, No. 18	drilled	⅜	⅝	1⅝	61.5	81.8	50.3	plate	46.6	75.8	92.7
DOUBLE-RIVETED SINGLE-WELT.											
20. Greig & Eyth, No. 20	drilled	⅜	⅝	2½	75.0	87.2	65.4	plate	55.6	74.1	85.0
SINGLE-RIVETED DOUBLE-WELT.											
21. Longridge, No. 26...	drilled	½	¾	2¼	66.7	117.8	78.6	plate	51.5	77.2	65.5
22. Do. No. 28...	punched	7/16	¾	2	62.5	161.6	101.0	plate	54.4	87.0	53.9
23. Do. No. 30...	drilled	7/16	¾	2	62.5	161.6	101.0	plate	58.5	93.6	57.9
24. Greig & Eyth, No. 22	drilled	⅜	⅝	1⅝	61.5	81.8	50.3	plate	62.7	102.0	62.3
DOUBLE-RIVETED DOUBLE-WELT.											
25. Longridge, No. 32...	punched	½	15/16	3	68.8	267.8	184.2	plate	67.0	97.4	36.4
26. Do. No. 38...	punched	7/16	13/16	2¾	70.4	244.6	172.2	plate	64.8	92.1	37.6
27. Brunel, Nos. 5 to 8..	punched	½	¾	4	81.2	110.0	89.3	plate, rivets	78.6	96.7	88.0
28. Greig & Eyth, No. 24	drilled	⅜	⅝	2½	75.0	174.4	130.8	plate, zigzag	63.5	84.7	48.6

breaking strengths are—52, 40.1, and 41.6 per cents. The strengths of Nos. 7 and 8 but slightly exceed those of the ½-inch plate-joints, Nos. 5 and 6: apparently in consequence of the near equality of the percentages of shearing sections—in the exact ratio, nearly; and though the fractures took place, in part, in the plates, evidently the shearing resistance, on the whole, was the measure of the strength. That the 7/16-inch joint, No. 2 of the table, should have realized 52 per cent strength, against 40.1 and 41.6 per cents for Nos. 7 and 8, with the same shearing section, points to the likelihood that the rivets of No. 2 were of stronger iron than those of the others.

For ⅜-inch plate-joints, single-riveted lap, of maximum strength, ¾-inch rivets at 2 inches of pitch are adopted.

Double-riveted Lap-joints in Iron Plates. Table No. 145.—The chain-riveted specimens, Nos. 12, 13, and 14, are scarcely comparable directly with zigzag riveting. The values are probably a little too high for boiler work. Besides, the pitches were not adjusted to the thicknesses. Again, the values in Nos. 15, 16, and 17 are probably too low for boiler work, since the rivets and their shearing sections are evidently of larger proportions than necessary. Rivets ¾ inch in diameter, at a pitch of 3 inches, for zigzag work, would have sufficed, whilst the net section would have been increased to 75 per cent. No. 18, it is true, a ⅜-inch joint, has 75 per cent of net section, and has only 64.6 per cent of breaking strength, against 72 per cent for No. 14. But the rivets are ⅝ inch in diameter, and the defect of so small a rivet appears to prevail in the double-riveted joint, as it did in the single-rivet joint. The following may be taken as the breaking strengths for the given sizes:—

For ⅜-inch plate...¾-inch rivets, 3 inches pitch, 70 per cent breaking strength.
For 7/16 „ ...¾ „ 3 „ 65 „ „
For ½ „ ...¾ „ 3 „ 60 „ „

Single-riveted Single-welt Joints in Iron Plates. Table No. 145.—There is but one instance of this joint, No. 19 in the table. For a ⅜-inch plate with ⅝-inch rivets, at 1⅝ inches of pitch, the breaking strength is only 46.4 per cent, which is less than that of the single-riveted lap-joint, No. 10 of identical sizes, 50.6 per cent. The inferiority of strength is readily explained by the nature of the joint, in which, when strained, the welt is bent in the middle, between the rows of rivets, in reverse directions,—following the bendings of the plates; and the line of stress is diverted to a greater extent, when passed into and through the welt than when passed through a direct lap-joint. This joint is therefore to be noted as the weakest form of riveted joint in iron plates; though it is redeemed in some degree when countersunk rivets are used. Against 56 per cent for the single-riveted lap-joint, a breaking strength of 50 per cent is adopted for this joint, in ⅜-inch plates, with ¾-inch rivets at 2 inches of pitch.

Double-riveted Single-welt Joints in Iron Plates. Table No. 145.—Similarly, this joint is weaker than the double-riveted lap-joint, as is manifested by the results in the table, of No. 20, a joint of this description having 55.6 per cent breaking strength, against 64.6 per cent in the case of No. 18, a double-riveted lap-joint of the same dimensions, making the difference 9 per cent. There is something exceptional in the amount of this difference. The breaking strength of the joint is taken at 65 per cent for ¾-inch rivets, at 3 inches of pitch.

Single-riveted Double-welt Joints in Iron Plates. Table No. 145.—The evidence, Nos. 21 to 24, is scanty. It points to the following as the probable strengths of joints for the given sizes:—

For ⅜-inch plate...¾-inch rivets, 2 inches pitch, 60 per cent breaking strength.
For 7/16 " ...¾ " 2⅛ " 55 " " "
For ½ " ...¾ " 2¼ " 52 " " "

Double-riveted Double-welt Joints in Iron Plates. Table No. 145.—The percentages of breaking strength of the three specimens, Nos. 25, 26, and 27 of the table, averaging 70.1 per cent, are not much less than the percentages of net plate-section, which average 73.5 per cent. In the same proportion, the breaking weight for ¾-inch rivets and 3 inches of pitch, making 75 per cent net section, is, say, 72 per cent.

The foregoing conclusions on the proportions and strength of riveted joints in iron plates are collected in a following table, page 671.

CHAP. IV.—ELEMENTAL STRENGTH OF STEAM BOILERS—RIVETED JOINTS (*continued*).

I. STRENGTH OF RIVETED JOINTS IN STEEL PLATES.—GREIG & EYTH.

The results of experiments on the strength of rivet-joints in steel plates, conducted by Messrs. David Greig and Max Eyth, Professor A. B. W. Kennedy, and Mr. C. H. Moberley, are here noticed, and the general comparative results are tabulated. The evidence in each case has been exhaustively analysed by the author, in his work on strength of materials, before referred to.

Messrs. David Greig and Max Eyth conducted a course of experiments on the strength of riveted joints in ⅜-inch boiler plates of iron and steel.[1]

The average results of preliminary tests of strength of plates and rivet-bars were, per square inch, as follows:—

	Per Square Inch. Yorkshire Iron.	 Mild Steel.
Tensile strength of plates, ⅜ inch thick, 2 inches wide..	22.27 tons.	25.83 tons.
Elongation in 6 inches of length; total length 10 inches	37 per cent.	28 per cent.
Tensile strength of rivet-bars, ⅝ inch in diameter, 30 inches in length...	22.23 tons.	28.83 tons.
Elongation in 10 inches of original length..................	27.9 per cent.	20.6 per cent.
Simple shearing strength of rivet-bars.........	19.01 tons.	22.18 tons.
Shearing strength, in parts of tensile strength of rivet-bars	85.5 per cent.	77 per cent.

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[1] See "Experiments referring to the use of Iron and Steel in High-pressure Boilers," in the *Proceedings of the Institution of Mechanical Engineers*, 1879; page 268.

The diameter of the ⅝-inch rivet-bars was reduced .03 inch, or $^{1}/_{32}$ inch by the shearing stress, in the direction of the pull, for both the iron and the steel.

The rivets were ⅝-inch rivets, inserted into drilled holes $^{11}/_{16}$ inch in diameter, and into punched holes, made with a punch $^{11}/_{16}$ inch in diameter, and a die $^{25}/_{32}$ inch in diameter, the die being $^{3}/_{32}$ inch larger than the punch. The conical punched hole was .708 inch in diameter at the top, and .790 inch at the bottom. The rivet-holes were elliptically elongated in the direction of the stress, after having supported the following maximum pressures:—

	Maximum Pressure.		Elongation.
Iron joints	15.7 tons.	......	.103 inch.
Steel joints	19.8 „	......	.225 „

Such elongation does not necessarily imply leakage as a consequence, as the indentation of the rivet forms a lap which tends to prevent leakage. At the same time the security against leakage is weakened.

Single rivets, actually formed, were tested for shearing resistance. Three steel plates, ⅜ inch thick, were riveted together by a ⅝-inch rivet set in an $^{11}/_{16}$-inch drilled hole, in double shear, making a total shearing section of .7424 square inch, the middle plate having been drawn out. The shearing stress, taking the average result for hand-riveting, hydraulic riveting and steam-riveting, per square inch of the double section, $^{11}/_{16}$ inch in diameter, was at the following rates:—

	Per Square Inch.	
	Iron.	Steel.
Shearing strength of rivet, in double shear	20.80 tons.	26.30 tons.
Excess over shearing strength of the simple rivet-bars	1.79 „	4.12 „
Excess in parts of „ „ „	9.4 p. cent.	18.5 p. cent.
Excess over tensile strength of plates	− 1.47 tons.	+ 0.47 ton.
Excess in parts of „ „	− 6.6 p. cent.	+ 1.8 p. cent.

The excess of the shearing resistance of the rivet over that of the simple bar is, no doubt, a measure of the frictional grip of the rivet; and it is shown in the last line above, that the total shearing resistance of the rivet, including grip-friction, does not, in the gross, differ very much from the tensile strength of the plate. But it is notable that the grip of the iron rivet, 1.79 tons, should be so much less than that of the steel rivet, 4.12 tons. This is explained, to some extent, by the fact of the greater tensile resistance of the steel rivet in contracting by cooling, in virtue of its greater strength; but chiefly by the less ductility of the steel rivet, that of the steel bars having been shown to be only three-fourths of that of the iron-bar.

The specimens of riveted joints were 6½ inches and 7½ inches in width. Those in iron were riveted with iron rivets, and those in steel with steel rivets. Slight differences in the thickness of the plates were reduced by calculation for a uniform thickness of ⅜ inch. All tests were made in duplicate, and the mean results are here given.

Ten specimens of single-rivet lap-joints, reckoned as Nos. 1 to 5, in duplicate, were made with two inches of lap, and four ⅝-inch rivets at 1⅝ inches of pitch, with ¹¹/₁₆-inch holes. They were tested for the relative values of drilled holes and punched holes. Four of these were of iron, two being drilled and two being punched, and all of them steam-riveted. In the punched specimens the smaller ends of the conical holes were in contact. The remaining six specimens were in steel, of which two had drilled holes, two were punched and then annealed, and two punched out and not annealed.

For the following tests the holes were drilled, and all steel plates were annealed.

Twelve single-rivet lap-joints, Nos. 6 to 11, were made in iron and in steel for the purpose of testing the comparative values of riveting by hand, by steam, and by hydraulic power. All the holes were drilled. They were of the same dimensions as Nos. 1 to 5 specimens.

The next four specimens, Nos. 12 and 13, of single-rivet lap-joints, were made in iron and in steel 7½ inches wide, with four ¾-inch rivets in drilled holes at 1⅞ inches of pitch; for comparison with the first specimens.

Four test-pieces, Nos. 14 and 15, were double-riveted zigzag lap-joints, made in iron and steel, 7½ inches wide, with 3 inches of lap, and with six ⅝-inch rivets, at a pitch of 2½ inches longitudinally, and 1⅝ inches diagonally, the longitudinal pitch-lines being 1 inch apart.

Four specimens of an exceptional double-riveted lap-joint, Nos. 16 and 17, were constructed 7½ inches wide, with 3¾ inches of lap, and eight ¾-inch rivets at 1⅞ inches pitch, longitudinally and diagonally, the longitudinal pitch-lines being 1½ inches apart.

Specimens of butt-joints, or welted joints, were next tested. Four specimens, Nos. 18 and 19, 6½ inches wide, were single-riveted single-welt joints, made with two inches of lap, with drilled holes and with four ⅝-inch rivets for each lap, at 1⅝ inches of pitch, like the single-riveted lap-joints. The welt or cover was ⅜ inch in thickness—the same as that of the plates.

Four specimens, Nos. 20 and 21, were double-riveted single-welt joints, 7½ inches wide, made with 3 inches of lap, having six ⅝-inch rivets at 2½ inches of pitch longitudinally, and 1⅝ inches diagonally, like the double-rivet lap-joints, Nos. 14 and 15.

Eight specimens, Nos. 22 to 25, of double-welt joints, were prepared, of which four were single-riveted and four were double-riveted, to the same dimensions as the single-welt joints, Nos. 18 to 21 respectively.

Mr. F. W. Webb, in discussion of the paper of Messrs. Greig and Eyth, stated that after having tried various proportions of rivets and pitches, he had arrived at a single-riveted double-welt joint for 7/₁₆-inch boiler plates of steel, possessing 71.6 per cent of the breaking strength of the whole plate, made with ¾-inch rivets at 2 inches of pitch. The covering plates are ⅜-inch thick and 5¼ inches wide, making a lap of 2⅝ inches on each plate, and a width of 1½ inches from the centre line of the rivets to the edges of the plates. He believed this distance, 1½ inches, to be the best; and previously to the adoption of the given proportions he had found that the holes went oval long before the joint ought to have been destroyed.

II. PROFESSOR A. B. W. KENNEDY'S EXPERIMENTS ON SINGLE-RIVETED LAP-JOINTS IN STEEL PLATES.

Professor Kennedy made a long course of experiments on single-riveted lap-joints in steel plates[1] of Landore S.S. quality, with a milder quality for the rivets. The plates were of three thicknesses—¼ inch, ⅜ inch, and ½ inch—and were tested for tensile strength in specimens having 10 inches of clear length. The ¼-inch pieces were tested of two different breadths, 1⅜ inches and 4 inches; the ⅜-inch pieces, of 2 inches and 3½ inches wide; and the ½-inch pieces, 4 inches wide. The influence of the differences of width was very slight. The exact sections of the pieces were taken.

Tensile Strength of Steel Plates.

Nominal Thickness.	Breaking Strength, pulled from Pins.		Breaking Strength, pulled in Wedge-grips.	
	Per Square Inch.	Extension in 10 Inches.	Per Square Inch.	Extension in 10 Inches.
inch.	tons.	per cent.	tons.	per cent.
¼	30.21	19.7	30.21	22.6
⅜	27.96	27.0	29.21	22.6
½	28.96	24.8	—	—

Sixteen specimens of rivet-steel were tested for tensile strength and shearing strength. They were turned down for testing from pieces of 11/16 inch, 15/16 inch, and 1 1/16 inches in diameter.

Strength of Rivet-steel.

Diameter.		Tensile Strength.		Shearing Strength (Single Shear).		
Original.	As Tested.	Per Square Inch.	Extension in 10 Inches.	Turned down to.	Per Square Inch.	In Parts of Tensile Strength.
inch.	inch.	tons.	per cent.	inch.	tons.	per cent.
11/16	.51	29.17	20.9	⅝	23.41	80.3
15/16	.62	30.73	19.9	⅝	26.63	86.7
1 1/16	.80	27.46	23.3	1	24.35	88.7
Averages..............		29.12	21.4	—	24.80	85.2

Specimens of plates, ¼ inch and ⅜ inch thick, and from 2 inches to 4 inches wide, were tested for the relative influence of punching and drilling rivet-holes, about 9/10 inch in diameter, at 2 inches of pitch. These tests were made in two series, with steels of different strengths:—

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[1] "Results of Experiments on Riveted Joints, made for the Institution of Mechanical Engineers" by Professor Alex. B. W. Kennedy, in the *Proceedings of the Institution of Mechanical Engineers*, 1881; page 205.

Tensile Strength of Plates Drilled and Punched, not Annealed.

Nominal Thickness.	Direct Tensile Strength of whole Plate, per Sq. Inch.	Tensile Strength Drilled.		Tensile Strength Punched.	
		Per Square Inch, Net Section.	In Parts of whole Plate.	Per Square Inch, Net Section.	In Parts of whole Plate.
inch.	tons.	tons.	per cent.	tons.	per cent.
HARDER STEEL.					
¼	34.41	38.17	110.7	34.74	101.2
⅜	31.45	35.15	111.9	34.14	108.4
MILDER STEEL.					
¼	29.00	32.26	111.2	30.87	106.4
⅜	28.87	32.00	110.8	30.53	105.8

These results show that the tensile strength per square inch of the net section in the line of the holes is greater than that of the whole plate per square inch:—consistently with the well ascertained behaviour of short specimens, to which the interspaces—between the perforations—are like.

Several specimens of the milder steel above tabulated were annealed. The result was a reduction of 3.3 per cent of tensile strength for the ¼-inch plates, and 3.6 per cent for the ⅜-inch plates; with 2 per cent less extension.

The punched holes were made with a ⅞-inch punch, and with dies successively 1/32 inch and 3/32 inch larger in diameter than the punch. The difference of clearance in the die did not make any sensible difference in the resisting strength of the plates.

Single-riveted lap-joints, in three series, were prepared for testing, hand-riveted. Each recorded result is the mean of the tests of two or three specimens.

The first series was made of ⅜-inch plate, containing two rivets in each. They were tested in pairs, with ¼-inch, ¾-inch, and 1-inch rivets respectively, for which the holes were drilled 1/32 inch larger in diameter. They were designed to break one of each pair in the plate, and the other in the rivets. The plates had an average of 29.97 tons per square inch tensile strength, with an average extension of 27 per cent in from 4 inches to 6 inches. The resistance was 4.4 per cent less when the pieces were annealed. The results of these tests are Nos. 1 to 6 in the following tablet.

The plates of each joint slipped on each other when the tensile stress reached the quantities undernoted; and the joints opened sufficiently to allow a slip of paper to be pushed in at the respective tensile stresses undernoted.

Single-riveted Lap-joint in ⅜-inch Steel Plate.

	Rivets.	Pitch.	Slipping at	Opened at
Nos. 1 and 2	½-inch.	.875 1.000	8½ tons per sq. inch.	13 tons per sq. inch.
No. 3............	¾ „	1.50	13½ „ „ „	8.8 „ „ „
No. 4............	¾ „	2.00	13½ „ „ „	5.5 „ „ „
No. 5............	1 „	2.25	8½ „ „ „	6.4 „ „ „
No. 6............	1 „	3.00	8½ „ „ „	3.3 „ „ „

As Professor Kennedy remarks, it is distinctly shown that the opening commences earlier as the pitch is increased.

The second series of tests was made on six joints of various laps and various pitches. They each had three ¾-inch rivets in drilled holes. The first three, Nos. 7 to 9, had varying laps, and therefore varying distances from the edge of the plate to the centre-line of the rivets; or, as Professor Kennedy distinguishes it, varying margins, or clear distance from the edge of the plate to the holes. The last three had the same lap, but different pitches. Strips tested for tensile strength broke with 29.9 tons per square inch, with 26.8 per cent extension in from 4 inches to 6 inches—nearly the same strength as that of the first series.

All the joints gave way by the shearing of the rivets.

The third series of six tests was made, of 18 specimens of single-riveted lap-joints, with drilled holes, each test being the average of three. The plates were nominally ⅜ inch and 5/16 inch thick, each specimen containing seven ¾-inch rivets of varying pitch. The strips tested for tensile strength, broke with the following average loads per square inch, and extensions in a length of 10 inches:—

⅜-inch plate...............	29.24 tons..................	22.6 per cent.
5/16-inch plate	29.64 "	25.1 "

The ¾-inch rivet-bar used for the joints in Nos. 13, 15, 17, and 18, was tested. One sample broke with a tensile stress of 29.03 tons per square inch. The shearing resistance of four samples of the bar, reduced to .621 inch in diameter, averaged 26.32 tons per square inch, or 90.7 per cent of the tensile strength of the single piece. This shearing resistance is greater than that of the rivet-bar used in the other specimens, which, as before stated, averaged a shearing strength of 24.80 tons per square inch.

Professor Kennedy states in discussion that the actual shearing resistance of the rivet in the plates, made from the same materials as those of the third series of tests, independently of the complication introduced by the bending of the plates, was ascertained by means of the shearing apparatus. It amounted to 27½ tons per square inch, which was $(27.5 - 26.33 =)$ 1.17 tons, or 4.4 per cent more than the normal resistance to shearing of the simple bar.

The plates forming the joints were observed to begin to slip on each other, when from 19 per cent to 27 per cent of the breaking load was applied, averaging 24 per cent.

III. MR. C. H. MOBERLEY'S EXPERIMENTS ON RIVETED JOINTS IN STEEL PLATES.

Mr. C. H. Moberley[1] made three series of tests of riveted joints of steel boiler plates, 9/16 inch, 7/16 inch, and 5/16 inch in thickness; of Landore S quality, with rivets of Landore rivet steel. The tests were made by Mr. David Kirkaldy.

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[1] See Mr. Moberley's papers, "Account of some Tests of Riveted Joints for Boiler Work;" and "Account of some Further Tests of Riveted Joints of Steel Plates for Boiler Work," in the *Proceedings of the Institution of Civil Engineers*, vol. lxix. 1881-82, page 337, and vol. lxxii. 1882-83, page 226.

9/16-*inch Plate-joints.*—The first series of tests was made with steel boiler plates, 9/16 inch in thickness nominally. The riveting was done by a hydraulic riveter, with an accumulator placed next to the machine. The pressure measured in the accumulator was 35 tons. The holes were punched; the plates were not annealed; the rivets were of steel. The joints were zigzag double-riveted double-welted; the inside welt or cover being 7/16 inch thick, and the outside welt 9/16 inch thick. The nominal diameter of the rivets was ¾ inch; and the diameter of the punch was 13/16 inch; the pitch in line, or longitudinally, was 3.775 inches; the pitch diagonally was 2.41 inches, the distance of the centre-lines of the two rows of rivets was 1.5 inches, and from the centre-line to the nearer edge of plate 1⅛ inch; the lap was 3.75 inches.

The joints tested as such, were of two forms. The first form, for specimens A, D, G, was 13.22 inches wide, and contained seven rivets in each lap. The second form, for specimens B, E, H, was 11.25 inches wide, and contained five whole rivets, and two half-rivets at the edges. The joints were made for the whole width of the specimens, as indicated in dot-lining, and caulked as if for boiler work; then slotted out.

Of the two groups of specimens, the first of each, A and B, was punched; in the second of each, D and E, the covers were punched, but the joint-plates were punched with a ⅝-inch punch, and drilled out to 13/16 inch in diameter; and in the third of each, G and H, the covers were punched, whilst the holes in the joint-plates were punched with a ⅝-inch punch, and drilled out to ¾ inch in diameter.

The ultimate tensile strength of the test pieces of plate cut from the six specimens averaged 30.3 tons per square inch, with an extension of 35.8 per cent in a length of 5 inches; the fractures were silky. The ultimate tensile strength of six ¾-inch steel rivets, turned down to ½ inch in diameter, was at the rate of, say, 29 tons per square inch, with a silky fracture. The shearing resistance was tested by making up three specimens, C, F, I, uniform with D, F, and H, and drilling out four rivets, leaving the three outer rivets at each side for shearing; with the following results:—

	Per Square Inch.
C, holes punched 13/16 inch....................................	23.8 tons.
F, holes punched ⅝ inch, drilled out to 13/16 inch.........	24.2 „
I, holes punched ⅝ inch, drilled out to ¾ inch...........	25.8 „
Average............	24.6 „

The average shearing resistance, 24.6 tons, is, say, 85 per cent of the tensile strength. This resistance is inclusive of that due to the frictional grip of the rivets.

To try the effect of drilling the rivet-holes in the joint-plates out of the solid, the covers being punched, two joints K, L, were prepared, in which $^{13}/_{16}$-inch holes were drilled for six rivets in each lap, arranged three in each row.

Excepting specimen D, all the specimens failed by fracture of the plate along the zigzag line through the rivets. Mr. Moberley, arguing that the two lines of rivets in each lap were too close, prepared four additional specimens, O, P, Q, and R, for testing, having six rivets in each lap, in two lines of three in each, 1⅝ inches apart, or ⅛ inch wider than in previous specimens. The pitch in line was the same as before, 3.775 inches, and it was 2.49 inches diagonally. The edge of the plate was 1¼ inches from the centre line of the rivets. These joints were made from like plates and rivets as for the previous joints. The mean breaking strength of O and P, was 28.9 tons per square inch, and that of Q and R was 29.7 tons.

The two joints, O and P, were made with $^{13}/_{16}$-inch drilled holes; and Q and R with ⅝-inch punched holes drilled out to $^{13}/_{16}$ inch; and the pressure for riveting was reduced to 25 tons. The fractures took place in the plates in the zigzag line of the rivets.

Four extra tests were made with single-riveted lap-joints, with the specific object of ascertaining the shearing resistance of 1-inch rivets, and the influence of the curving of the plates, as for the circular seams of a boiler, upon the shearing resistance. Two flat specimens, S and T, and two specimens, M and N, curved to a diameter of 10 feet 3 inches, were prepared. They were 14.2 inches wide, having each five 1-inch rivets, at 2.84 inches of pitch, with 3.05 inches of lap. Specimen rivets turned down to .714 inch in diameter, were tested for tensile strength, and broke with an average of 28.3 tons per square inch. In each of the four specimens, the five rivets were shorn; with the mean stress, 23.29 tons per square inch for the flat joints, and 24 tons for the curved joints. In each case, the plates were bent at the joint, in the usual way. On the ultimate tensile strength, 28.3 tons per square inch, those shearing resistances were 82.3 per cent for the flat joint, and 84.8 per cent for the curved joint. Obviously, the second pair of joints were to some extent fortified by the curved form of the plates increasing the resistance to bending, and hence the greater percentage of resistance to shearing.

IV. DEDUCTIONS FROM THE EVIDENCE ON STEEL PLATE RIVETED JOINTS.

A selection of characteristic results of the tests just noticed, is collected in table No. 146, following, comprising results of experiments made by Messrs. Greig & Eyth, Professor Kennedy, and Mr. Moberley. For the sake of comparison, column 14 is added, showing the shearing resistance of rivets per square inch, ascertained by special tests, free from bending stress, in parts of the tensile strength of the whole plate per square inch. In columns 15, 16, and 17, are given the spacing, or the distance apart of the two rows of rivets, with the diagonal pitch, for double-riveted joints.

Table No. 146.—SUMMARY RESULTS OF CHARACTERISTIC TESTS OF RIVETED JOINTS OF STEEL PLATES. 1879–83.
(Experiments of Greig & Eyth, Kennedy, and Moberley.)

JOINT, AND REFERENCE.	Drilled or Punched.	Thickness of Plate.		Rivets.		Net Sec. of Plate, in Parts of that of whole Plate.	Shearing Section.		Fracture in Plate or in Rivets.	Breaking Strength in Parts of that of whole Plate.	Breaking Strength per Sq. In. in parts of that of whole Plate per Sq. In.		Approximate Shearing Resistance of Rivets per Sq. In., in parts of Tensile Strength of Plate per Sq. In.	Double-riveting.		
		Nominal.	Actual.	Diameter.	Pitch.		In Parts of Net Section of Plate.	In Parts of Section of whole Plate.			of Net Plate Section.	of Shearing Section.		Two Rows of Rivets apart, or Pitch of Rows.	Diagonal Pitch.	
															Ins.	In Parts of Longl. Pitch.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
SINGLE-RIVETED LAP-JOINTS.		inch.	inch.	inch.	inches.	pr.ct.	pr. ct.	pr. ct		pr. ct.	per ct.	per ct.	per ct.	ins.	ins.	pr. ct.
1. Greig, Nos. 3 and 10	drilled	⅜	.375 or ⅜	⅝	1⅝ or 1.625	61.5	81.8	50.3	rivets	57.6	93.6	114.3	102.0	—	—	—
2. Kennedy, No. 3	drilled	⅜	.379	¾	1.50	50.0	155.0	77.5	plate	55.0	110.0	71.0	90.0	—	—	—
3. Kennedy, Nos. 9 and 13	drilled	⅜	.390	¾	1.55	51.6	143.0	73.8	plate, rivets	57.8	112.0	78.3	91.7	—	—	—
4. Kennedy, No. 5	drilled	⅜	.384	1	2.25	55.5	164.0	91.0	rivets	53.5	96.4	58.8	—	—	—	—
5. Kennedy, No. 18	drilled	5/16	.347	¾	1.77	57.1	129.0	74.1	rivets	60.8	106.5	82.6	93.4	—	—	—
DOUBLE-RIVETED LAP-JOINTS.																
6. Kennedy, Nos. 25 and 26	drilled	¾	.736	1	3.10	67.7	101.6	68.8	plates straight, rivets	69.0	102.0	100.3	—	1.9	2.45	79.0
7. Moberley, E, F, No. 12	punched	7/16	.44	¾	2.75	72.7	100.4	73.0	plates zigzag and straight, rivets	66.1	90.9	90.6	91.2	1.56	2.083	75.7
8. Greig, No. 15	drilled	⅜	.375 or ⅜	⅝	2.50	75.0	87.2	65.4	rivets	70.0	93.3	107.0	102.0	1	1.625	65.0
9. Kennedy, Nos. 19 and 20	drilled	⅜	.376	¾	2.90	74.1	109.3	81.0	plate straight, rivets	80.0	108.0	98.8	—	1.6	2.15	74.1
10. Moberley, C, D, No. 14	punched	5/16	.32	⅝	2.50	75.0	102.3	76.7	plate straight, rivets	74.15	98.9	96.7	95.5	1	1.625	65.0
SINGLE-RIVETED SINGLE-WELT.																
11. Greig, No. 19	drilled	⅜	.375 or ⅜	⅝	1⅝ or 1.625	61.5	81.8	50.3	rivets	55.8	90.7	110.9	102.0	—	—	—
DOUBLE-RIVETED SINGLE-WELT.																
12. Greig, No. 21	drilled	⅜	.375 or ⅜	⅝	2.50	75.0	87.2	65.4	rivets	68.0	90.7	104.0	102.0	1	1.625	65.0
SINGLE-RIVETED DOUBLE-WELT.																
13. Webb	—	7/16	—	¾	2	62.5	160.6	100.4	plate	71.6	114.6	71.4	—	—	—	—
14. Greig, No. 23	drilled	⅜	.375 or ⅜	⅝	1⅝ or 1.625	61.5	163.6	100.6	plate	60.4	98.2	60.0	102.0	—	—	—
DOUBLE-RIVETED DOUBLE-WELT.																
15. Kennedy, Nos. 28 and 29	drilled	¾	.739	1	4.40	77.3	125.0	96.6	plate straight	69.4	89.8	71.8	—	2.3	3.18	72.3
16. Moberley, K, L, No. 7	drilled	9/16	.56	¾	3.775	80.1	104.3	83.5	plate zigzag, rivets	67.75	84.5	81.1	79.5	1.5	2.41	63.0
17. Moberley, O, P, No. 8	drilled	9/16	.55	¾	3.775	80.1	104.3	83.5	plate zigzag	73.15	91.3	85.6	103.1	1.625	2.49	66.0
18. Greig, No. 25	drilled	⅜	.375 or ⅜	⅝	2.50	75.0	174.4	130.8	plate zigzag	68.0	90.7	52.0	102.0	1	1.625	65.0
19. Kennedy, Nos. 22 and 23	drilled	⅜	.393	.65	2.75	76.4	160.6	122.7	plate straight	80.7	105.7	65.8	—	1.45	2.00	72.7

Single-riveted Lap-joints of Steel Plates, Nos. 1 *to* 5. Table No. 146.—Of these joints, Nos. 1 and 3 are equal in breaking strength, having 57.6 per cent and 57.8 per cent respectively. But the net sections of plate are 61.5 per cent and 51.6 per cent, showing 10 per cent difference in favour of No. 1, which has not been effective in adding to the strength of the joint; and consequently, whilst the breaking strength per square inch of the net plate section, column 12, is only 93.6 per cent of that of the whole plate for No. 1, it is 112.0 per cent for No. 3. Accordingly, No. 1 broke in the rivets, leaving a margin of unutilized strength in the plates, whilst No. 3 broke in the plates as well as in the rivets; and it may be said that the rivets in No. 1 were as strong as the plates as well as the rivets in No. 3. True, the shearing resistance of the rivets per square inch was the greater in No. 1, being 102 per cent of the breaking strength of the whole plate, against 91.7 per cent in No. 3; but the difference is more than compensated by the larger proportion of shearing section in No. 3, which is 73.8 per cent of the section of the whole plate, against only 50.3 per cent in No. 1, or nearly one-half more; and, naturally, whilst the breaking strength per square inch of the shearing section of No. 3 is only 78.3 per cent of that of the whole plate, it is 114.3 per cent for No. 1.

Here is a correspondence between the relative breaking strengths of the joints per square inch of shearing section, column 13, and the shearing resistance by special tests, column 14. That is to say, in both cases, No. 1 shows the greater percentage of efficiency; and is enabled, with the less proportion of rivet section, to realize as high a breaking strength as No. 3.

But strong though No. 1 is, there is, as has been shown, a want of balance or of equal resistance, the net plate being more resistant than the rivets; and a stronger joint could be made by reducing the net plate section a little, and increasing the shearing section. This has been effected in No. 5 joint, which has 57.1 per cent of net section of plate and 74.1 per cent of shearing section. The augmentation of shearing section, no doubt, is apparently much greater than the reduction of net plate section; but it includes compensation for the reduction of shearing resistance of the rivets per square inch, column 14, from 102.0 per cent to 93.4 per cent of the tensile strength of the whole plate.

The adjustment or compensation here effected in No. 5 joint, compared with No. 1, is effected by the enlargement of the rivets to ¾ inch in diameter, and the increase of the pitch to 1.77 inches. The less thickness of No. 5, too, is in favour of an augmentation of the percentage of breaking strength.

Double-riveted Lap-joints of Steel Plates, Nos. 6 *to* 10. Table No. 146.—In a double-riveted joint, the tensile stress between the two rows of rivets is distributed between the two plates which form the lap. If not equally divided between them, it must be nearly so. In consequence, the stress in each plate between the rows of rivets is only half of that in the single plates beyond the rivets; that is, the average stress, for, no doubt, the stress in the plates is variable both longitudinally and transversely, being localized about the rivets: reaching partly in zigzag lines from rivet to rivet, and partly in lines from each rivet at right angles to the joint, traversing the corresponding triangular space. It may be presumed, therefore, looking to the stress in each plate independently, that the nearer row of rivets takes by far the greater proportion of that longitudinal stress; and that the farther row takes much the less proportion of that stress. There is a similar distribution of force with respect to the other plate forming the joint. The observations of Mr. Moberley, on the unequal elongation of the several rows of bolt holes by means of which his specimens were attached to the testing machine, lend countenance to such assumption. It follows from this view of the distribution of stress, that the middle portion of plate between two rivets in the front row may be comparatively unsupported by the intermediate rivet in the back row, if the distance apart of the two rows is excessive, and may be in a state of suspension between the two front rivets, between which the portion of plate between the two rows of rivets may be wedged by the longitudinal stress. It would then become excessively strained, and would be liable to fracture before the plate would give way at the rivets. The strain would be shown by the curving outwards of straight lines drawn on the surface of the plates parallel to the joint. Professor Kennedy shows similar distortions of test strips under tensile stress,[1] and such distortion under the circumstances above noted must be a source of weakness.

The results of the tests, Nos. 6 to 10, show a breaking strength of joint ranging from 66.1 per cent to 80 per cent of that of the whole plate. Comparing Nos. 6 and 9, having ¾-inch and ⅜-inch plates, with large allowance of diagonal pitch, and fractures by shearing the rivets, as well as by tension through the plates straight, the breaking strengths are 69 per cent and 80 per cent respectively. These strengths are almost exactly in the ratio of the shearing sections, which are 68.8 per cent and 81 per cent of the sections of the whole plate, and are thus directly provided for, on the part of the rivets. The difference of strength is also, in part, represented by the difference of the net sections, which are 67.7 per cent and 74.1 per cent. But there remains a comparative deficiency of resistance in the plates of No. 6; and it suggests the inquiry whether the diagonal pitch was not too great in No. 6. It was 2.45 inches, and not unlikely, on the assumption of unequally distributed stress already noticed, the plate was strained unequally—more at and between the nearer line of rivets, and less at the further line. It happens that of all the double-riveted lap-joints brought into table No. 146, No. 6 was made with the widest pitches longitudinally and diagonally, 3.10 inches and 2.45 inches, and that the diagonal pitch bore the greatest proportion—79 per cent—to the longitudinal pitch. The plate was also the thickest of the group; but mere thickness would not affect the inequality of drag and of distribution of stress. It is, therefore, probable that the diagonal pitch of No. 6, which was 2.45 inches, or

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[1] See his paper of 1881, already referred to, in the *Proceedings of the Institution of Mechanical Engineers*, 1881; plate 31.

79 per cent of the longitudinal pitch, was too much for securing a maximum of resistance; and that the joint would have been stronger if the diagonal pitch had been less. The stocks of Mr. Moberley's specimens were occupied by 1-inch pins—of the same diameter as the rivets in No. 6, placed at a pitch longitudinally of 3¾ inches, and diagonally 3 inches, which is 80 per cent of the longitudinal pitch—about the same proportion, as it happens, as that in No. 6 test; and under such similar conditions it is probable that each half of the joint depended for the greater part on the nearer line of rivets for its active resistance to fracture.

The discrepancy between the pitches of Nos. 6 and 9 joints is made directly evident, in comparing the longitudinal pitches, which are nearly equal—being 3.10 inches and 2.90 inches, differing by 7 per cent—with the spacing apart of the two rows, which are 1.9 inches and 1.6 inches, differing by 20 per cent. The tensile stress would only in a much diminished proportion reach the farther rivets of the ¾-inch plate-joints.

It is probable, moreover, that the greater thickness of No. 6 induced a greater degree of bending stress, and of weakening in consequence, than in No. 9.

Comparing No. 9, having 80 per cent of strength, with No. 8, having 70 per cent, in which the thicknesses of plate are identical, and the net sections are nearly equal, the shearing section of No. 8 is much less than that of No. 9,—being 65.4 per cent against 81 per cent. Evidently, the inferiority of strength of No. 8 results from the comparative deficiency of shearing section, in consequence of which the joint fractured only in the rivets.

No. 10, also, is stronger than No. 8, evidently because there is a greater percentage of shearing section,—76.7 per cent against 65.4 per cent of the whole plate, although the size of rivets and the pitchings are identical. The difference of percentage arises from the difference of thickness of the plates, which are ⅜ inch and 5/16 inch in thickness, in consequence of which the rivets are relatively stronger for the thinner plate-joint, No. 10. There is, besides, the less tendency of the thinner plates of No. 10 to bend, in favour of No. 10.

Comparing, again, No. 10 with No. 9, No. 10, having 74.15 per cent of breaking strength, approaches nearer in strength than any other joint in the group to No. 9, which has 80 per cent. These percentages of strength correspond nearly to the respective shearing sections, 76.7 per cent and 81 per cent of that of the whole plate; and, seeing that both joints broke through the rivets as well as through the plates, and were joints of equal resistance, the strengths were practically regulated by the shearing sections. It should be added as an additional reason for the greater breaking strength of No. 9, that the plates were less ductile than those of No. 10, the extension of test-samples of No. 9 plates having been only 18.8 per cent in 10 inches, against 27.2 per cent for No. 10.

No. 7 test, of a 7/16-inch plate-joint, shows the lowest strength, 66.1 per cent, in the group, although its proportions of rivets and pitches are nearly like those of No. 9, which has 80 per cent of strength. The difference is due to the greater thickness of No. 7 joint-plates, and the greater bending stress in consequence.

Single-riveted Single-welt Joints in Steel Plates, No. 11, table No. 146.—The specimen is identical in dimensions with No. 1, the single-riveted lap-joint, and had 55.8 per cent breaking strength—about 2 per cent less than that of the simple lap-joint.

It may be taken that the single-riveted single-welt joint is not quite so strong as the single-riveted lap-joint.

Double-riveted Single-welt Joint in Steel Plates, No. 12, table No. 146.—The specimen is identical in dimensions with No. 8, and had the breaking strength 68 per cent—being, as was found for the preceding case, 2 per cent less than that of the plain double-riveted lap-joint, No. 8.

It may be taken that the double-riveted single-welt joint is not quite so strong as the double-riveted lap-joint.

Single-riveted Double-welt Joint in Steel Plates, Nos. 13 and 14 tests, table No. 146, shows 60.4 per cent breaking strength. The breaking strengths of the three single-riveted joints of identical dimensions, of the same steel, and by the same constructors, Messrs. Greig and Eyth, are here brought together for comparison. Plate ⅜ inch, rivets ⅝ inch, pitch 1⅝ inches:—

Single-riveted Joint, ⅜-inch Steel Plate.	Fracture.	Breaking Strength.
No. 1. Single-riveted Lap-joint	rivets	57.6 per cent
No. 11. Single-riveted Single-welt	rivets	55.8 „
No. 14. Single-riveted Double-welt	plate	60.4 „

The double-welted joint, breaking through the plate, clearly shows the greatest percentage of breaking strength. But the particular value here found is to be adjusted for ¾-inch rivets at 2 inches of pitch. The test No. 13, by Mr. Webb, shows for these dimensions 71.6 per cent breaking weight. To harmonize with the general evidence for the other joints a mean of 65 per cent is taken as the breaking strength of the single-riveted double-welt joint in steel.

Double-riveted Double-welt Joint in Steel Plates, Nos. 15 to 19, table No. 146.—In these, as in the tests of double-riveted lap-joints, Professor Kennedy's joint in ⅜-inch plate, No. 19, has proved of the greatest efficiency, having 80.7 per cent breaking strength. The rivets were in double-shear, and all the fractures took place in the plates. Comparing the ¾-inch plate-joint, No. 15, with No. 19, their respective strengths are nearly identical with those of the corresponding double-riveted lap-joints, Nos. 6 and 9, being 69.4 per cent and 80.7 per cent, in double-welts, against 69 per cent and 80 per cent when simply double-riveted. Comparing the two joints of ¾-inch plates, Nos. 6 and 15, the rivets are in both 1 inch in diameter, but they are placed at a wider pitch, 4.40 inches, in No. 15, against 3.10 inches in No. 6, and the net plate section is increased by 14 per cent, from 67.7 per cent to 77.3 per cent. In face of this greater net plate section, how does it happen that the breaking strength through the plate was not increased; with the advantage, too, of a straight pull between welts? The only answer appears to be that, for reasons already presented, the diagonal pitch is too long. It is 3.18 inches, or 72.3 per cent of 4.40 inches—the longitudinal pitch—and the pitch of the two rows of rivets is 2.3 inches; whence it appears to have followed that the back row of rivets co-operated to a very limited extent in supporting the plate, and that the plate was overstrained and fractured between the widely pitched rivets of the front row, in advance of the portions of the plate nearer to the rivets; that it was thus broken in detail, and failed to exhibit all its resistance simultaneously.

No. 19, having a slightly greater breaking strength—80.7 per cent against 80 per cent for No. 9—has also a slightly greater net section of plate, 76.4 per cent against 74.1 per cent: but the rivets and the pitch are smaller, being .65 inch and 2.75 inches against ¾ inch and 2.90 inches. Consistently, the diagonal pitch is a little less for No. 19, being 2 inches against 2.15 inches for No. 9, and having 1.45 inches between the rows of rivets against 1.6 inches for No. 9.

The three intermediate tests, 16, 17, and 18, gave way by zigzag fracture in the plate. Nos. 16 and 17, of 9/16-inch plates, are identical in dimensions, except that in No. 17 the rows of rivets are wider apart than in No. 16, the spacing being 1.625 inches against 1.5 inches. The extra width in No. 17 was clearly beneficial, as it increased the breaking strength from 67.75 per cent to 73.15 per cent, although the first object of augmenting the spacing—to cause fracture in the straight line—was not attained. Mr. Moberley considers that the width between the rows of rivets might probably be increased to 1¾ inches, with a slight addition of strength to the joint.

No. 17, having 9/16-inch plates, and 3.775 inches of pitch, is the most nearly like No. 15, of ¾-inch plate, and having 4.40 inches of pitch. It has about 3 per cent more net section, but the principal difference consists in the diagonal pitch, which is only 2.49 inches against 3.18 inches in No. 15, whilst the two rows are only 1.625 inches apart, against 2.3 inches. The proportions of No. 17 tell in its favour, by exhibiting 73.15 per cent of breaking strength against 69.4 per cent by No. 15.

But No. 16, as was noticed, had only 67.75 per cent breaking strength, a little less than No. 15 had. The only apparent reason of the difference is that the plates of No. 16 broke zigzag and the rivets were shorn, whilst in No. 15 the plate broke straight, and there was a greater shearing section than in No. 16. It appears that No. 15 erred as much by excessive spacing, or distance between the rows of rivets, as No. 16 erred by insufficient spacing.

No. 18, of ⅜-inch plate, making the third of the zigzag fractures, has nearly the same net section, 75 per cent, as the first of them, No. 16, of 9/16-inch plate, namely 80.1 per cent; and has the same breaking strength, 68 per cent. With considerably different pitches—2.50 inches and 3.775

inches—the percentages of the diagonal pitches are nearly the same, 65 per cent and 63 per cent, both of which are deficient in length.

Again, No. 18 is more nearly like No. 19, having ⅜-inch plates, and other dimensions nearly alike. The net sections also are alike—75 per cent and 76.4 per cent. Yet, whilst No. 18 had only 68 per cent of breaking strength, No. 19 had 80.7 per cent. The gain appears to be solely attributable to the longer diagonal pitch of No. 19, which was 2 inches, against 1.625 inches in No. 18.

The breaking strengths of the three double-riveted joints of identical dimensions, of the same steel, constructed by the same engineers, Messrs. Greig & Eyth, are subjoined, for comparison:—plate ⅜ inch, rivets ⅝ inch, pitch 2.50 inches, diagonal pitch 1.625 inches.

	Double-riveted Joint, ⅜-inch Steel Plate.	Fracture.	Breaking Strength.
No. 8.	Double-riveted lap-joint...............	rivets	70 per cent.
No. 12.	Double-riveted single-welt............	rivets	68 „
No. 18.	Double-riveted double-welt...........	plate, zigzag	68 „

These results, though they do not show maximum efficiency, are remarkable for the nearness to equality of the breaking strengths. They make an instructive contrast with the corresponding values for single-riveted joints by the same constructors, table No. 146. The single-riveting, freed from bending stress in the double-welt, showed a decidedly greater breaking strength than in the lap or the single-welt joint; whereas, the double-riveting, when similarly guarded from bending stress in the double-welt, shows no more breaking strength than the lap or the single-welt joint,—an argument that double-riveted joints, in ⅜-inch steel plates, are not materially weakened by bending stress.

CHAP. V.—ELEMENTAL STRENGTH OF STEAM BOILERS—RIVETED JOINTS (*continued*).

STANDARD JOINTS IN IRON AND STEEL PLATES.

The results of No. 5 test, slightly modified for ⅜-inch plates, are taken as a basis for standard proportions of single-riveted joints in steel plates, of equal resistance. Assuming that the tensile strength of the rivet-bars is at least equal to that of the plates, the pitch is taken as 2 inches, with ¾-inch rivets, making the net section five-eighths or 62.5 per cent of the whole plate section, which is a slight advance on the net section of No. 5, without overdrawing on the resistance of the rivets, the section of which is 94 per cent of that of the net section, and is 58.75 per cent of the whole plate section. The breaking strength is fixed at 60 per cent.

For double-riveted lap-joints in ⅜-inch steel plates, of equal resistance, the proportions of joint No. 9, table No. 146, are taken as a basis for standard proportions. The pitch is extended to 3 inches, by which the net section is slightly increased without unduly diminishing the proportion of shearing resistance with ¾-inch rivets. The diagonal pitch is taken as 2¼ inches, being 75 per cent of the longitudinal pitch. The breaking strength is 80 per cent.

For single-riveted single-welt joints, in ⅜-inch plates, the proportions of the single-riveted lap-joint are adopted, with 58 per cent breaking strength.

For double-riveted single-welt joints, in ⅜-inch plates, the proportions of the double-riveted lap-joint are adopted, with 78 per cent breaking strength.

For single-riveted double-welt joints, in ⅜-inch plates, the proportions for the single-riveted lap-joint are adopted, with 65 per cent breaking strength.

For double-riveted double-welted joints, in ⅜-inch plates, of equal resistance, the proportions of joint No. 19 are adopted as a basis. For the sake of uniformity, the pitch is increased to 3 inches, and the rivets to ¾ inch in diameter, as in the double-riveted joint; and the breaking strength is taken as 80 per cent. A smaller rivet, in double shear, may no doubt suffice for staying power, as in No. 19, where it is .65 inch in diameter, but the ¾-inch rivet is preferred, not only for the sake of uniformity, but for that of stability of the plate-bearing on the rivet.

The conclusions arrived at in the preceding sections on the proportions and strength of riveted joints in boiler-plates of iron and of steel, ⅜ inch thick, with the percentages of net section and shearing section, and of breaking strength in terms of these sections, are collected in the following table, No. 147:—

Table No. 147.—RIVETED JOINTS IN BOILER PLATES OF IRON OR STEEL, ⅜ INCH THICK, OF EQUAL RESISTANCE AND MAXIMUM STRENGTH:—Standard Dimensions and Proportions.

JOINT IN ⅜-INCH PLATE.	Thickness of Plate.	Rivets.		Net Section of Plate, in parts of that of whole Plate.	Shearing Section.		Breaking Strength, in parts of that of whole Plate.		Breaking Strength, per Square Inch, in parts of that of whole Plate per Square Inch.			
		Diameter.	Pitch Longitudinally.		In parts of Net Section of Plate.	In parts of whole Plate.			Of Net Plate Section.		Of Shearing Section.	
							Iron.	Steel	Iron.	Steel.	Iron.	Steel.
1	2	3	4	5	6	7	8	9	10	11	12	13
	inch.	inch.	ins.	per cent.	per cent.	per cent.	per cent.	per cent.	per cent.	per cent.	per cent.	per cent.
Single-riveted lap..............	⅜	¾	2	62.5	94.0	58.75	56	60	89.6	96.0	95.3	102.1
Single-riveted single-welt.....	⅜	¾	2	62.5	94.0	58.75	50	58	80.0	92.8	85.1	98.7
Single-riveted double-welt...	⅜	¾	2	62.5	188.0	117.50	60	65	96.0	104.0	51.1	55.3
Double-riveted lap..............	⅜	¾	3	75	104.7	78.50	70	80	93.3	106.7	89.1	102.0
Double-riveted single-welt.....	⅜	¾	3	75	104.7	78.50	65	78	86.7	104.0	82.8	99.3
Double-riveted double-welt...	⅜	¾	3	75	209.4	157.00	72	80	96.0	106.7	45.9	51.0

NOTES TO TABLE.—1. The diagonal pitch for the double-riveted joints is 2¼ inches, or 75 per cent of the longitudinal pitch. It is equal to the length of interspace between rivets longitudinally.

2. The distance apart of the two rows of rivets is 1.68 inches,—say 1 11/16 inches bare,—or 56 per cent of the longitudinal pitch.

The relations of thickness of plate, diameter of rivets, and pitch of rivets, here shown for ⅜-inch plate-joints, are applicable to other thicknesses of plates, and are generalized in the following table, No. 148. Proportions for lap are included, allowing a "margin," or width of plate outside the rivet equal to the diameter of the rivet, which is according to the prevailing practice.[1] The "spacing" denotes the distance apart of the centre-lines of the two rows of rivets in double-riveting.

Table No. 148.—Standard Proportions of Riveted Joints in Iron and Steel.

Thickness of plates...............	=	unity
Diameter of rivets.................	=	thickness of plate × 2
Pitch of rivets (single-riveting)	=	thickness of plate × 5⅓ diameter of rivets × 2⅔
Pitch of rivets (double-riveting)	=	thickness of plate × 8 diameter of rivets × 4
Diagonal pitch (double-riveting)	=	longitudinal pitch × ¾ diameter of rivets × 3
"Spacing" (double-riveting) ...	=	longitudinal pitch × .56, or 9/16
Lap (single-riveting)..............	=	thickness of plate × 6 diameter of rivets × 3
Lap (double-riveting)............	=	thickness of plate × 10.48, or 10½ diameter of rivets × 5.24, or 5¼

On these proportions, the diameter and pitches of rivets suitable for plates of from ⅛ inch to 11/16 inch in thickness, are as given in the following table, No. 149. The calculation is not extended for thicknesses greater than 11/16 inch, for which the corresponding rivet has a diameter of 1⅜ inches, as this size is assumed to be the maximum properly available for practical operations. For thicknesses of plate greater than 11/16 inch, therefore, the standard proportions are not available.

The percentages of strength given in table No. 147 for single-riveted lap and single-riveted single-welt joints hold only for plates ⅜ inch thick. For thinner plates the percentages are greater, and for thicker plates they are less, than the actual strengths respectively. These variations have already been exemplified in the preceding discussion on iron-plate joints. But single-riveting, as above defined, is not now practised except in situations where the margin of tensile resistance is ample, as, for instance, in the transverse seams of cylindrical boilers.

Such calculations can only be expected to give approximate results, considering the variety of conditions under which joints are made; and although equal resistance, that is, equality of the resistances of the net section and the shearing section, may be secured by suitably adjusting to each other the diameter and the pitching of the rivets, it is arguable that the larger the rivets and the wider the interspaces, the greater is the breaking strength of the joint, seeing that the percentage of net section is thus augmented. But other considerations have to be weighed. Upon the whole, the proportions given in the table, No. 149, are considered to combine most effectively the conditions of equal resistance, maximum strength, convenience of construction, handiness, and tightness under steam.

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[1] See Appendix I., on Riveted Joints, to Professor Kennedy's paper already referred to. See foot-note, page 659.

Table No. 149.—STANDARD RIVETED JOINTS OF MAXIMUM STRENGTH IN IRON AND STEEL PLATES OF VARIOUS THICKNESS.

Net section 62.5 per cent of whole plate section.

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets.				Lap.	
		Single-riveting.	Double-riveting.			Single-riveting.	Double-riveting.
			Longitudinal.	Diagonal.	Spacing.		
inch.	inches.	inches.	inches.	inches.	inches.	inches.	inches.
1/8	1/4	2/3	1	3/4	9/16	3/4	1 5/16
3/16	3/8	1	1 1/2	1 1/8	27/32	1 1/8	2
1/4	1/2	1 1/3	2	1 1/2	1 1/8	1 1/2	2 5/8
5/16	5/8	1 2/3	2 1/2	1 7/8	1 13/32	1 7/8	3 1/4
3/8	3/4	2	3	2 1/4	1 11/16	2 1/4	3 15/16
7/16	7/8	2 1/3	3 1/2	2 5/8	2	2 5/8	4 5/8
1/2	1	2 2/3	4	3	2 1/4	3	5 1/4
9/16	1 1/8	3	4 1/2	3 3/8	2 1/2	3 3/8	5 7/8
5/8	1 1/4	3 1/3	5	3 3/4	2 13/16	3 3/4	6 1/2
11/16	1 3/8	3 2/3	5 1/2	4 1/8	3 1/16	4 1/8	7 1/4

NOTE TO TABLE.—The standard proportions are not applicable for joints of plates exceeding 11/16 inch in thickness, in consequence of the unfeasibility of using rivets of greater diameter than 1 3/8 inches.

Whilst the proportions just set forth for riveted joints are considered to be the best proportions—providing a special size and pitch of rivets for each thickness of plate—other sizes and pitches may of course be adopted, resulting in different degrees of strength of joint. At the same time, they may be designed to form joints of equal resistance, and for this purpose formulas may be constructed for calculating the pitch of rivets of any given diameter for plates of any given thickness; or conversely, the diameter of rivets for given pitches and thicknesses. The double-welted joints, which hold the rivets in double shear, and in which shearing section is thus provided in excess, are not strictly joints of equal resistance; and the rivets might consequently be made smaller in diameter—augmenting the interspace and the resisting net section correspondingly—the shearing section at the same time being sufficient. The augmentation of resisting area that might thus be gained would not be considerable, and, moreover, though rivets of a diameter of 5/8 inch, for instance, are tolerable in a joint of 3/8-inch steel plate, the evidence is unfavourable to their use in iron-plate joints. Let—

t = thickness of the plate, in inches.
d = diameter of the rivets, in inches.
p = pitch of the rivets, in inches.

a = area of net plate section, between two rivets in the straight line, in square inches.

a' = shearing section of one rivet.

r = ratio of the shearing section of the rivets to the net section of the plate, or $\frac{a'}{a}$.

The area of the net section is equal to $(p-d)t$; it is also equal to $\frac{.7854\,d^2}{r}$. Therefore, $\frac{.7854\,d^2}{r}=(p-d)\,t$; and $\frac{.7854\,d^2}{r\,t}=p-d$; whence—

Pitch of Rivets for equal resistance.

$$p=\frac{.7854}{r\,t}d^2+d \text{..................................} \quad (1)$$

giving the value of the pitch p in terms of the diameter of the rivets and the thickness of the plate.

Resolving the equation (1) for the value of d, the diameter of the rivets, the general expression is obtained.[1]

Diameter of Rivets for equal resistance.

$$d=\sqrt{\frac{r\,t}{.7854}p+\left(\frac{r\,t}{1.57}\right)^2}-\frac{r\,t}{1.57} \text{.....................} \quad (2)$$

To reduce the formulas to their simplest forms as adapted to each kind of joint:—

Single-riveted Lap-joint.—The shearing section in table No. 147 is 94 per cent of the net plate section, and the value of the ratio r is .94. Substitute this value in equations (1) and (2), and they become by reduction,—

$$p=.835\frac{d^2}{t}+d \text{...} \quad (3)$$

$$d=\sqrt{1.20\,t\,p+.36\,t^2}-.60\,t \text{..........................} \quad (4)$$

These formulas are applicable also for single-riveted single-welt and double-welt joints. For the double-welt joint, in which the rivets are in double shear, it is understood that only half the shearing section, representing one rivet, is calculated with, or 94 per cent of the net plate section, as for the lap-joint.

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[1] The formula (1) is an affected quadratic equation, reduced thus:—Divide both sides by the coefficient of d^2, and, $d^2+\frac{r\,t}{.7854}d=\frac{r\,t}{.7854}p$; add the square of half the coefficient of d, then—

$$d^2+\frac{r\,t}{.7854}d+\left(\frac{r\,t}{1.57}\right)^2=\frac{r\,t}{.7854}p+\left(\frac{r\,t}{1.57}\right)^2.$$

Take the square root of both sides:—

$$\sqrt{d^2+\frac{r\,t}{.7854}d+\left(\frac{r\,t}{1.57}\right)^2}=\left(d+\frac{r\,t}{1.57}\right)=\sqrt{\frac{r\,t}{.7854}p+\left(\frac{r\,t}{1.57}\right)^2};$$

whence the value of d, given in the text, equation (2),—

$$d=\sqrt{\frac{r\,t}{.7854}p+\left(\frac{r\,t}{1.57}\right)^2}-\frac{r\,t}{1.57}.$$

Double-riveted Lap-joint.—The shearing section in table No. 147 is 104.7 per cent of the net plate section, for two lines of rivets; or it is 52.35 per cent for the front line; and $r = .5235$. By substitution in formulas (1) and (2), and reduction,—

$$p = 1.5\frac{d^2}{t} + d \qquad (5)$$

$$d = \sqrt{\frac{2}{3}tp + \frac{t^2}{9}} - \frac{t}{3} \qquad (6)$$

These formulas are applicable for double-riveted single-welt and double-welt joints. In the case of the double-welt joints, half the total shearing section, which is 209.4 per cent of the net section, is 104.7 per cent for the front line of rivets. As these are in double shear, one half of this percentage, or 52.35 per cent, the same as for the lap-joint, is taken for calculation.

To verify these formulas by examples, calculate the pitch and the diameter of rivets for ⅜-inch plate-joints. For the single-riveted lap-joint, in ⅜-inch plate, the diameter of the rivet is ¾ inch; to find the pitch, $t = .375$, and $d = .75$; and by substitution in formula (3),—

$$p = .835\frac{.75^2}{.375} + .75 = 1.251 + .75 = 2 \text{ inches},$$

as in the table. Again, to find the diameter of rivet, when the pitch is given, $t = .375$, and $p = 2$; and by substitution in formula (4),—

$$\begin{aligned} d &= \sqrt{(1.20 \times .375 \times 2) + (.36 \times .375^2)} - (.60 \times .375) \\ &= \sqrt{.90 + .0506} - .225 \\ &= \sqrt{.9506} - .225 = .975 - .225 = .75 \text{ inch}, \end{aligned}$$

as in the table.

For the double-riveted lap-joint, in ⅜-inch plate, similarly, to find the pitch in the straight line. The thickness $t = .375$, and the diameter $d = .75$; then, by substitution in formula (5),—

$$p = 1.5\frac{.75^2}{.375} + .75 = 2.2499 + .75 = 2.9999 \text{ or } 3 \text{ inches},$$

the pitch, as in the table. To find the diameter, the pitch $p = 3$, and $t = .375$, and substituting these values in formula (6),—

$$\begin{aligned} d &= \sqrt{\left(\frac{2}{3} \times .375 \times 3\right) + \left(\frac{.375}{3}\right)^2} - \frac{.375}{3} \\ &= \sqrt{.75 + .0156} - .125 \\ &= \sqrt{.7656} - .125 = .875 - .125 = .75 \text{ inch}, \end{aligned}$$

the diameter of the rivets, as in the table.

In the table, No. 150, of riveted joints of equal resistance, the pitches for rivets of various diameters are given for various thicknesses of plates, fulfilling, in each instance, the condition of equal resistance, and, of course, of maximum strength, for each size of rivet in plate-joints of a given thickness.

The thickness of plates, column 1, varies from ¼ inch to 1⅛ inches. The diameters of rivets, column 2, for each thickness of plate, comprehend the widest diversities of current practice, and they embrace, of course, the standard joints already given in table No. 149. For reasons already explained, the maximum diameter of rivets entered in the table is 1⅜ inches. The respective pitches, in single-riveting and double-riveting, are given in duplicate,—decimally and fractionally,—in columns 3, 4, 5, and 6. The percentages of net plate section, for single-riveting and double-riveting, are given in columns 7 and 8; and these are nominally measures of the relative strengths of the joints.

Table No. 150.—RIVETED JOINTS OF EQUAL RESISTANCE, IN IRON AND IN STEEL PLATES:—MACHINE-RIVETED.

Tensile Strength of the Rivets at least equal to that of the Plates.

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets.				Net Plate Section in Parts of whole Section.	
		Single-riveting.		Double-riveting.		Single-riveting.	Double-riveting.
1	2	3	4	5	6	7	8
inch.	inches.	inches.	inches.	inches.	inches.	per cent.	per cent.
¼	½	1.333	1 11/32	2.00	2	62.5	75.0
	9/16	1.619	1⅝	2.462	2 15/32	65.3	77.1
	⅝	1.930	1 15/16	2.969	2 31/32	67.6	78.9
5/16	9/16	1.408	1 13/32	2.081	2 3/32	60.0	73.0
	⅝	1.667	1 21/32	2.500	2½	62.5	75.0
	11/16	1.950	1 15/16	2.940	2 15/16	64.8	76.6
	¾	2.253	2¼	3.450	3 7/16	66.7	78.3
⅜	⅝	1.495	1½	2.188	2 3/16	58.2	71.4
	11/16	1.740	1¾	2.585	2 19/32	60.5	73.4
	¾	2.000	2	3.000	3	62.5	75.0
	13/16	2.282	2 9/32	3.453	3 15/32	64.4	76.5
	⅞	2.580	2 19/32	3.937	3 15/16	66.1	77.8
7/16	11/16	1.590	1 19/32	2.308	2 5/16	56.7	70.2
	¾	1.824	1 13/16	2.678	2 11/16	58.9	72.0
	13/16	2.072	2 1/16	3.076	3 1/16	60.8	73.6
	⅞	2.333	2 11/32	3.500	3½	62.5	75.0
	15/16	2.615	2⅝	3.941	3 15/16	64.1	76.2
	1	2.908	2 29/32	4.429	4 7/16	65.6	77.1
½	¾	1.689	1 11/16	2.438	2 7/16	55.6	69.2
	13/16	1.915	1 29/32	2.793	2 25/32	57.6	70.9
	⅞	2.154	2 5/32	3.172	3 3/16	59.4	72.4
	15/16	2.405	2 13/32	3.574	3 9/16	61.0	73.8
	1	2.667	2 11/16	4.000	4	62.5	75.0
	1 1/16	2.947	2 15/16	4.449	4 7/16	64.0	76.1
	1⅛	3.238	2¼	4.922	4 15/16	65.3	77.2

Table No. 150 (*continued*).

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets.				Net Plate Section in Parts of whole Section.	
		Single-riveting.		Double-riveting.		Single-riveting.	Double-riveting.
1	2	3	4	5	6	7	8
inch.	inches.	inches.	inches.	inches.	inches.	per cent.	per cent.
9/16	¾	1.585	1 19/32	2.250	2¼	52.7	66.7
	13/16	1.792	1 25/32	2.573	2 9/16	54.7	68.4
	⅞	2.011	2	2.916	2 29/32	56.5	70.0
	15/16	2.242	2¼	3.281	3 9/32	58.2	71.4
	1	2.484	2½	3.667	3 21/32	59.7	72.7
	1 1/16	2.738	2¾	4.072	4 1/16	61.2	73.9
	1⅛	3.000	3	4.500	4½	62.5	75.0
	1 3/16	3.279	3 9/32	4.945	4 15/16	63.8	76.0
	1¼	3.569	3 9/16	5.416	5 13/32	65.0	76.9
⅝	⅞	1.898	1 29/32	2.713	2 23/32	53.9	67.7
	15/16	2.111	2⅛	3.046	3 1/32	55.6	69.2
	1	2.336	2 11/32	3.400	3 13/32	57.2	70.6
	1 1/16	2.571	2 9/16	3.771	3 25/32	58.7	71.8
	1⅛	2.816	2 13/16	4.163	4 5/32	60.0	73.0
	1 3/16	3.070	3 1/16	4.568	4 9/16	61.3	74.0
	1¼	3.333	3 11/32	5.000	5	62.5	75.0
11/16	15/16	2.005	2	2.854	2 27/32	53.2	67.2
	1	2.215	2 7/32	3.182	3 3/16	54.9	68.6
	1 1/16	2.433	2 7/16	3.525	3 17/32	56.3	69.9
	1⅛	2.662	2 21/32	3.886	3⅞	57.7	71.0
	1 3/16	2.898	2 29/32	4.262	4¼	59.0	72.1
	1¼	3.148	3 5/32	4.660	4 21/32	60.3	73.2
	1 5/16	3.404	3 13/32	5.069	5 1/16	61.4	74.1
	1⅜	3.667	3 11/16	5.500	5½	62.5	75.0
¾	1	2.113	2⅛	2.999	3	52.7	66.7
	1 1/16	2.319	2 5/16	3.319	3 5/16	54.2	68.0
	1⅛	2.534	2 17/32	3.657	3 21/32	55.6	69.2
	1 3/16	2.756	2¾	4.006	4	56.9	70.4
	1¼	2.990	3	4.375	4⅜	58.2	71.3
	1 5/16	3.230	3 7/32	4.756	4¾	59.4	72.4
	1⅜	3.475	3 15/32	5.156	5 5/32	60.4	73.5
13/16	1	2.028	2 1/32	2.845	2 27/32	50.7	64.8
	1 1/16	2.222	2 7/32	3.146	3 5/32	52.2	66.2
	1⅛	2.426	2 7/16	3.462	3 15/32	53.6	67.5
	1 3/16	2.635	2⅝	3.788	3 25/32	55.0	68.7
	1¼	2.856	2 27/32	4.135	4⅛	56.2	69.8
	1 5/16	3.082	3 3/32	4.492	4½	57.4	70.8
	1⅜	3.318	3 5/16	4.866	4⅞	58.6	71.7

Table No. 150 (*continued*).

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets.				Net Plate Section in Parts of whole Section.	
		Single-riveting.		Double-riveting.		Single-riveting.	Double-riveting.
1	2	3	4	5	6	7	8
inch.	inches.	inches.	inches.	inches.	inches.	per cent.	per cent.
⅞	1	1.954	1 31/32	2.715	2 23/32	48.8	63.2
	1 1/16	2.140	2⅛	2.997	3	50.3	64.6
	1⅛	2.333	2 11/32	3.295	3 9/32	51.8	65.9
	1 3/16	2.532	2 17/32	3.602	3 19/32	53.1	67.0
	1¼	2.741	2¾	3.928	3 15/16	54.4	68.2
	1 5/16	2.955	2 31/32	4.264	4¼	55.6	69.2
	1⅜	3.179	3 3/16	4.616	4⅝	56.8	70.2
15/16	1⅛	2.252	2¼	3.150	3 5/32	50.0	64.3
	1 3/16	2.442	2 7/16	3.442	3 7/16	51.4	65.5
	1¼	2.641	2 21/32	3.750	3¾	52.7	66.7
	1 5/16	2.846	2 27/32	4.068	4 1/16	53.9	67.7
	1⅜	3.059	3 1/16	4.401	4 13/32	55.0	68.8
1	1⅛	2.182	2 3/16	3.024	3 1/32	48.4	62.8
	1 3/16	2.364	2⅜	3.301	3 5/16	49.8	64.0
	1¼	2.555	2 9/16	3.594	3 19/32	51.1	65.2
	1 5/16	2.750	2¾	3.895	3 29/32	52.3	66.3
	1⅜	2.953	2 31/32	4.212	4 7/32	53.4	67.4
1 1/16	1 3/16	2.294	2 9/32	3.176	3 3/16	48.2	62.6
	1¼	2.478	2 15/32	3.455	3 15/32	49.6	63.8
	1 5/16	2.665	2 21/32	3.743	3¾	50.8	65.0
	1⅜	2.860	2⅞	4.045	4 1/32	51.9	66.0
1⅛	1 3/16	2.233	2¼	3.065	3 1/16	46.8	61.3
	1¼	2.410	2 13/32	3.334	3 11/32	48.1	62.5
	1 5/16	2.590	2 19/32	3.609	3 19/32	49.3	63.6
	1⅜	2.778	2 25/32	3.895	3 29/32	50.5	64.7

NOTES TO TABLE.—1. The lap for each joint is equal, for single-riveting, to 3 times the diameter of the rivets; and for double-riveting, to 5¼ times the diameter.

2. The diagonal pitch in double-riveting is ¾ths of the longitudinal pitch; and the spacing is 9/16ths of the longitudinal pitch.

It is shown in the table that, for a given thickness of plate, the percentage of net section—nominally the percentage of strength—varies with the diameter of the rivets:—the larger the rivets the greater the net section. As the thickness of plate advances, the percentages of net section and strength range lower. For instance,—

	Single-riveting.	Double-riveting.
⅜-inch plate, maximum net section……	66.1 per cent……	77.8 per cent.
¾ „ do. ……	60.4 „ ……	73.5 „
1⅛ „ do. ……	50.5 „ ……	64.7 „

This is not all. The actual strength of single-riveted joints, it has been shown experimentally, falls off much more rapidly with the increasing thickness, than in the ratio of the net section; and those of double-riveted joints also fall off, though less rapidly.

CHAP. VI.—ELEMENTAL STRENGTH OF STEAM BOILERS—RIVETED JOINTS (*continued*).

Experimental Breaking Strength of Riveted Joints in Relation to Thickness.

From the evidence for iron plate-joints it appears that single-riveted joints in ½-inch plates had only 40 per cent breaking strength against 56 per cent for ⅜-inch plates. For 7/16-inch plate-joints, it may be taken at 50 per cent.

IRON.	
½-inch plate-joints......................	40 per cent breaking strength.
7/16 „ „	50 „ „
⅜ „ „	56 „ „

In steel, the evidence shows inferentially that a 16-inch plate-joint was stronger than the ⅜-inch standard joint. Thus:—

STEEL.	Net Section.	Breaking Strength.
Standard joint ⅜-inch plate (table No. 147)...	62.5 per cent.......	60 per cent.
5/16-inch plate-joint..................................	57.1 „	.... 60.8 „

With less net section the 5/16-inch steel plate-joint had a little more breaking strength than the ⅜-inch plate.

With respect to double-riveted lap-joints, the drift of evidence, which is rather mixed, tends to show that the breaking strength is less unfavourably affected by thickness in steel joints than in iron joints.

Breaking Joint.

Mr. Longridge records the results of tests for the increase of resistance obtained by breaking joint, the distance between the seams being 12 inches; from which he concludes that there was a gain of 14 per cent strength, for ½-inch plate-joints, single-riveted lap.

Chain-riveting.

The few tests recorded of chain-riveted joints indicate,—in the welted joints, at least, which are not given in table No. 145,—a slightly greater strength of joint than the tests of zigzag joints.

Resistance of Perforated Iron Plates to Compression at Rivet-holes.

The resistance of a perforated plate to the compressive action of a rivet upon the inside of the rivet-hole, is a factor which need not enter into calculations for the strength of rivet-joints; for the power of resisting compression, at the side of a rivet-hole filled by a rivet, and firmly inclosed between the head and the point of the rivet, is usually much in excess of the compressive force that can actually be applied. Mr. Tangye's experiments[1] show clearly that the resistance of an iron plate may, under like circumstances, amount to a pressure of 40 tons per square inch, before the metal yields to any material extent; and that the elastic resistance amounted to 20 tons per square inch. The section of a plate exposed to compression at a rivet-hole is to be reckoned as the product of the diameter, d, of the rivet by the thickness, t, of the plate; whilst the section of rivet opposed to it is $.7854d^2$. Taking the respective resistances, to compression and to shearing, as 40 tons and 20 tons per square inch, or as 2 to 1, the resistance to compression is at least as much as the resistance to shearing, when the thickness is such that $2td = .7854d^2$, or $2t = .7854d$, or, in round numbers, $t = \frac{2}{5}d$. In all suitably proportioned joints, the thickness of the plate is greater than two-fifths of the diameter of the rivet; and, therefore, even when the compressive resistance of the plate is limited, for calculation, to 40 tons per square inch, it is greater in amount than the shearing resistance of the rivet.

Steam-tightness of Riveted Joints.

Plate-joints may be strong enough; they must be steam-tight also. The effort of steam to slip through between the plates at a seam, is exerted in bending apart or opening the plates between two rivets; and the deflection is probably regulated by the laws which apply to the deflection of beams under transverse stress. The deflection should vary as the pressure, as the cube of the pitch, and inversely as the cube of the thickness. If, therefore, the pitch and the thickness be increased in equal proportions, the joint should be equally well tight, under equal pressures; and, if the pressure be increased, the shortening of the pitch, to provide the needful resistance to leakage, may be proportionally much less than the increase of pressure. For instance, supposing the thickness remains the same, if the pressure be doubled, the pitch may only be reduced in the ratio of $\sqrt[3]{2}$ to 1, or as 5 to 4.

CHAPTER VII.—TESTS OF THE STRENGTH OF STEAM BOILERS.

Lancashire Boiler.—In applying the hydraulic test to boilers of the construction and dimensions of the ordinary Lancashire boiler, it is the practice to carry the pressure to 150 lbs. per square inch, and to strain fine cords across the flat ends, to act as datum lines from which to gauge the ends at, say, twelve points. Mr. L. E. Fletcher states that it is found as a rule that the plate under pressure bulges outwards at the centre from $^1/_{16}$ inch to $\frac{1}{8}$ inch, and on the removal of the pressure returns to its original position without suffering any permanent set. In experimental hydraulic bursting tests, the ends, $\frac{1}{2}$ inch thick, of the boiler, as above described, have stood repeatedly a pressure of 275 lbs. per square inch, without leakage or any appearance of distress; but, under a pressure of 300 lbs., the front end-plate displayed signs of weakness about the mud-hole, beneath the furnace-tubes. With this exception, the greatest bulging was ⅛ inch at the front and 3/16 inch at the back, with a maximum permanent set of 1/16 inch.

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[1] See *A Manual of Rules, Tables and Data*, page 582.

A wrought-iron boiler of this type, 21 feet long, 7 feet in diameter, with two 2¾-feet furnace-tubes, was tested by hydraulic pressure by Mr. Lavington E. Fletcher, for the Manchester Steam Users' Association. It was constructed to the drawings of the association, of Snedhill plates, and tested, in 1874–76, at the works of Mr. T. Beeley, Hyde Junction. The shell was of 7/16-inch plates, the tubes of ⅜-inch plates, and the ends of ½-inch plates. The shell and tubes were made with seven belts of plating; the belts in the shell being each in three lengths circumferentially. The double-riveting was made at 2.44 inches pitch in line, and at 1¾ inches pitch diagonally, on 3⅜ inches of lap. The rivets were slightly double-conical, 25/32 inch in diameter at the middle, having .48 square inches of sectional area—90 per cent of the section of the entire plate, and 135 per cent of the net section of plate in line. The belts in the tubes were of one plate circumferentially, welded at the longitudinal joints, and strengthened with flange seams. The flat ends were each in one piece. The longitudinal stay-bolts were removed. The rivet-holes in the shell were punched; those of the tubes were drilled. Test-pieces, consisting of strip-joints, double-riveted and single-riveted, and plain strips, were afterwards cut out of the boiler, and tested in a testing machine.

Under hydraulic test, all the rents occurred longitudinally in the shell, starting either at one of the openings cut in the plating for the manhole or fittings, or at a longitudinal seam. The circumferential seams, the ends, and the tubes, were uninjured. A longitudinal seam double-riveted by machine gave way at a pressure of 310 lbs. per square inch; the rent extending longitudinally for the entire length of the seam, 3 feet, and nearly one foot beyond into the next plates at the ends of the seam. The equivalent tensile strength was 13.34 tons per square inch of section of the entire plate. The net section of plate between the rivets was two-thirds of the section of the continuous plate. Then $13.34 \times \frac{3}{2} = 20.01$ tons per square inch—the ultimate tensile resistance between the rivet-holes. A longitudinal seam double-riveted by hand, burst for its length of 3 feet under a pressure of 300 lbs. per square inch, the rent extending more than 2 feet into the next plate at one end of the seam. A longitudinal seam single-riveted by hand was burst along the seam, without tearing into the neighbouring whole plates, under a pressure of 275 lbs. per square inch. A cast-iron manhole mouthpiece, 17 inches in diameter inside and 1 inch thick, was burst by a pressure of 200 lbs. per square inch. The upper part was blown entirely off, the fracture passing round the cylindrical part of the casting. A cast-iron base, 5 inches in diameter and 1 inch thick, for a blow-off tap, was burst in the same manner by a pressure of 295 lbs.; and a cast-iron base for a 6-inch steam-junction valve, ⅞ inch thick, gave way under 275 lbs. pressure, a piece having been blown out at one side. An oval manhole, 17 inches by 13 inches, unguarded, that is unfortified by an extra ring of metal, failed at a pressure of 200 lbs. by the fracture of the plate at one end of the hole, in consequence of the cover being forced outwards by the pressure. A wrought-iron manhole mouthpiece, 16 inches in diameter, sustained a pressure of 300 lbs. per square inch without showing any signs of yielding. Under hydraulic pressure, with a dome of 7/16-inch plate, 3 feet in diameter, on a 7-feet shell, the whole of the shell-plate at its base being cut away so as to form an opening as large as itself, the flange at the base of the dome ripped at a pressure of 150 lbs. per square inch. At another trial with a dome of the same size, and a portion only of the shell-plate cut away, the straining and leakage were so much that not more than 235 lbs. pressure could be obtained.

The direct tensile strength of the boiler plate was proved to average 20.6 tons per square inch, which is about the same as the resistance of the metal between the rivets of the double-riveted joints, when in place in the boiler shell. But the separate samples of double-riveted joints showed, when tested by Mr. Kirkaldy, only 16.22 tons resistance per square inch of net section. Whence it appears, as Mr. Fletcher points out, that the belt formed by the double thickness of plate at the circumferential seam of rivets contributed to the relative augmentation of resistance, in place in the boiler; and that a breaking of joint of the longitudinal seams is beneficial in the same sense.

Mr. Fletcher, writing in December, 1888, remarks that these tests showed that the strength of the metal between the rivets in the actual boiler, though the holes were punched, might be taken as practically equal to the strength of the plates per square inch of section; that, at a working pressure of 75 lbs. per square inch, the factor of safety, 4, was obtained when the boilers were fitted with wrought-iron manhole mouthpiece and fitting branches; that ½-inch end-plates, without the aid of longitudinal stay-bolts, had ample strength; that the flanged seams were adequate to give ample strength to the furnace-tubes; and that these tubes, though only ⅜ inch thick, were stronger than the shell, at all events at a cold hydraulic test; that they confirmed the view that the stress at the longitudinal seams of rivets in the shell, is much greater than at the ring seams, the boiler being drop-tight at most of the ring seams of rivets, while at the longitudinal seams before rupture it leaked profusely; that the cast-iron mountings were untrustworthy, and that these bursting experiments helped to break through the practice of adopting cast-iron fittings.[1]

Messrs. John Elder & Co.'s Experiments.[2]—A cylindrical boiler, 11 feet 3 inches in diameter inside, of ¾-inch plates, double-riveted, was burst by a hydraulic pressure of 230 lbs. per square inch, equivalent, by formula (3), page 620, to 9.375 tons per square inch of the continuous section of plate. The rivets were $1^1/_{16}$ inches in diameter, at a lineal pitch of $2\frac{3}{4}$ inches, leaving 61.36 per cent of solid metal between. The shearing section of the rivets amounted to 86 per cent of the entire section of the plate, and to 141 per cent of the net section of the metal. The breaking strength of the metal between the rivets was $\left(9.375 \times \frac{100}{61.36} = \right)$ 15.31 tons per square inch.

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[1] The fractured pieces are exhibited in the Museum of the Manchester Steam Users' Association.

[2] Described in *Engineering*, July 21, 1876; page 47.

A cylindrical superheater, 99.915 inches in diameter inside, of $^9/_{16}$-inch plates, double-riveted with $^{13}/_{16}$-inch rivet-holes, at $2\frac{1}{2}$ inches pitch, was burst, at the line of rivets, by a hydraulic pressure of 245 lbs. per square inch, equivalent, by formula (3), page 620, to a bursting pressure of 9.988 tons per square inch of the entire plate, and 14.775 tons per square inch of the net section between the rivets; the net section having been 67.6 per cent of the entire section. The shearing section of two rows of rivets was 77.2 per cent of the entire section; and 114.2 per cent of the net section.

Now, the tensile strength of the plates of the boiler and of the superheater, tested by Mr. Kirkaldy, averaged 20.5 tons and 20.2 tons respectively, per square inch. It thus appears that the net tensile resistance of the plates between the rivet-holes, was only,—

For the boiler $\left(\frac{15.31 \times 100}{20.5} = \right)$ 74.7% of the direct strength.

For the superheater $\left(\frac{14.775 \times 100}{20.2} = \right)$ 73.1% of the direct strength.

Showing that the plates had lost one-fourth of their direct power of resistance.

The three experimental results of testing for the bursting strength of two boilers and a superheater, just noticed, supply evidence of the considerable augmentation of the strength of the rivet-joints of boiler-plates, when well placed, breaking joint, in a boiler, relatively to their normal strength as tested in the form of flat specimens. They also confirm the deduction that the thicker plates had the weaker joints proportionally.

Messrs. Greig & Eyth made some tests of the strength of boiler-shells, and of complete boilers for portable engines.[1] Three barrels were constructed, 2 feet $11\frac{1}{2}$ inches in diameter inside, and 3 feet long, of $\frac{3}{8}$-inch plates, machine-riveted. Two of them, one of iron and one of steel, were made with single-riveted longitudinal lap-seams, punched holes, annealed for steel, and $\frac{5}{8}$-inch rivets at $1\frac{5}{8}$ inches of pitch. The third barrel was of steel, double-riveted longitudinally, with drilled holes and $\frac{5}{8}$-inch rivets. The barrels were tested by hydraulic pressure, and they all gave way by violent leakage at the longitudinal seams, under pressures much below those due to the breaking strength of the joints:—the result of compression of the rivets, and elongation of the rivet-holes.

The first barrel, of iron, leaked violently when the pressure reached 420 lbs. per square inch; and, by caulking, it was made to bear 450 lbs.

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[1] See foot-note, page 656.

pressure. It was increased in diameter, under the stress, by .038 inch. The stress at this time did not exceed 74 per cent of the breaking strength at the net section of plate.

The second barrel, of steel, permitted a maximum pressure of 490 lbs. per square inch; equivalent to 69 per cent of the breaking strength at the net section of plate.

The third barrel, of steel, double-riveted, admitted 650 lbs. pressure per square inch, being about 69 per cent of the breaking strength of the net section.

It appeared that the shearing stress borne by an ordinary ⅝-inch steel rivet, before starting to leak hopelessly, was about 14,500 lbs., or 6½ tons, being from 66 per cent to 69 per cent of the shearing strength of the rivet.

Two complete boilers, of the locomotive form, were constructed, one of iron, and one of mild steel. The barrels were single-riveted longitudinally as well as transversely, except at the junction with the fire-box shell, where the circular seam was double-riveted. The barrels were of ⅜-inch plate, 2 feet 11½ inches in diameter, like the separate barrels before tested, and 8 feet long, made of two rings of plates 4 feet long. The boilers supported a maximum pressure of 420 lbs. per square inch in the iron boiler; and in the steel boiler, with ordinary caulking, 520 lbs., and with caulking by sledge-hammers, 620 lbs. The leakage by the longitudinal seams prevented greater rise of pressures. There was neither rupture nor permanent set in any part of either boiler. The diameter of the barrel increased only by .015 inch over the circular seam; only by .144 inch across the middle of the plate. The circular seams of the boiler gave no trouble whatever.

It is seen that the barrel segments, 3 feet long, closed at the ends, bore higher pressures than the boiler-barrels, probably because they were stayed by the ends, and were of shorter lengths. The pressures in the boilers were too low to cause, in any part, either fracture or set. The bulging of the barrel between the transverse rivet-seams, was evidenced by the unequal enlargements of the diameter. Mr. L. E. Fletcher had observed a similar bulging between circular seams in his experimental boiler.

CHAPTER VIII.—STEAM BOILERS—DEFINITIONS.

Steam boilers of every kind are constructed of parts common to all, though varied in form to suit special conditions.

The *shell* of a boiler, as the name signifies, is the outer inclosure of iron or of steel. It may be spherical, cylindrical, or rectangular in figure; or it may have a combination of these forms.

The *dome*, or the *steam-chest*, on the upper side of the boiler, is a reservoir into which the steam is collected as it is generated, and whence it is drawn to supply the engine by the *steam-pipe*, which is fitted with a *stop-valve*.

The *furnace*, or *fireplace*, is the chamber in which the fuel is burnt for the production of heat. When the furnace is in the form of a tube within the shell, it is the *furnace-tube;* when it is of a flat-sided or cubical form, it is the *fire-box.*

The *flues,* or conduits for the products of combustion, are either *external* to the shell, or they consist of *internal* cylindrical metal flues of comparatively small diameter, in considerable number. Such circular flues are called *flue-tubes,* and they constitute a *multitubular flue.* The flue-tubes are fixed at the ends into *tube-plates* by *ferules;* or they may be *expanded* against the tube-plate.

The *manhole* is the entrance to the boiler, for inspection, cleaning out, and repair.

Mudholes are placed at or near the bottom, for the discharge of sediment and for washing out the boiler.

The water is supplied to the boiler by the *feed-apparatus:* a *feed-pump,* or an *injector,* or an *inspirator.* Its level is indicated by a *float.*

The boiler is emptied through the *blow-off cock.*

The surface of the water is cleared by the *scum-cock.*

Brine-pumps may be used instead of blow-off cocks, to draw off the brine from marine boilers.

Sediment-collectors receive the solid impurities in the water.

Surplus steam escapes by the *safety-valves.*

By *vacuum-valves,* air is admitted into the boiler when the pressure is less than that of the atmosphere.

Fusible plugs are inserted in the crown of the furnace or fireplace, which are melted, and give vent to the steam when the crown-plates become overheated by any cause.

The *pressure-gauge* indicates the pressure in the boiler.

The *glass water-gauge* shows the level of the water in the boiler. So also do the *gauge-cocks,* approximately.

The boiler is strengthened by *stays,* which may consist of *rods, bolts,* or *gussets.*

The boiler is covered with *clothing* or *cleading;* or by *non-conducting* material.

The *fire-grate* carries the fuel, and consists of *fire-bars, furnace-bars,* or *grate-bars,* supported by *cross-bearers* or *bar-frames.*

The *mouthpiece* is the entrance to the furnace, and it rests on the *dead-plate,* which is the floor or sole of the entrance.

The *fire-door* is fitted to and hung to the mouthpiece or to the *furnace-front,* which is applied to ordinary flue-boilers, being usually of cast iron, or wrought iron, fastened to the end of the boiler, or to brickwork in front of the boiler.

CHAPTER IX.—LANCASHIRE STEAM BOILERS.

It is estimated, on good authority, that there are upwards of 100,000 steam boilers at work in the United Kingdom. The so-called "Lancashire," or double-flue cylindrical boiler, may be accepted as the standard type of boiler. It has been arrived at gradually. The globular or spherical shape was very early adopted as one of greatest capacity, and as a shape which was maintained by the pressure, without any tendency to distortion. But spherical boilers presented a minimum of surface through which the heat of the burning fuel was to pass, and it was followed by the upright cylindrical form, like a bottle, in which the surface was sensibly developed; then by "hay-stack" boilers and "waggon" boilers. But these, in departing from the balanced forms of the spherical and the cylindrical boilers, required to be stayed internally; and were adapted only for a pressure of 7 lbs. per square inch above the atmospheric pressure. A reaction took place. The long cylindrical boiler, with semi-spherical ends, was adopted, without any other metal than the shell of the boiler itself for strength. It is laid with its axis horizontal, or nearly horizontal, and below it, at one end, is placed the fire, inclosed by brickwork as usual. Thus, a great development of heating surface was effected, the hot gaseous products being conducted in flues round the boiler before being discharged into the chimney. A boiler of this type, in use at ironworks and collieries, is illustrated by figs. 264 and 265. It is 6 feet in diameter and 34½ feet long, made of ½-inch iron plates, for a working pressure of 80 lbs. per square inch. It is mounted with two safety-valves; a feed-pipe and valve; a steam-pipe and stop-valve, connected to the main pipe; a glass water-gauge, and three test-taps and pipes, passed through the crown of the boiler. The boiler slopes slightly downwards towards the far end, where the blow-out pipe is connected to the bottom of the boiler, whence it is extended underground to the front, where the blow-out tap is connected, through which the water is discharged.

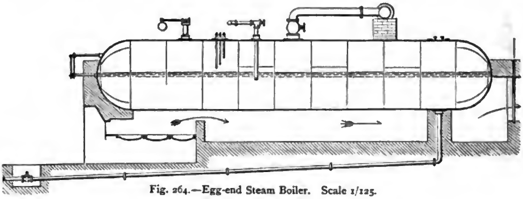
Fig. 264.—Egg-end Steam Boiler. Scale 1/125.

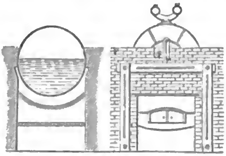
Fig. 265.—Egg-end Boiler. End View and Section. Scale 1/125.

In the coal and iron districts of the North of England, there are such boilers, 45 feet long and 4 feet in diameter. For utilizing the otherwise wasted gases, a length of 60 feet or even 80 feet, and a diameter of 4½ feet, are frequent.

Besides the egg-end cylindrical boiler, just described, numerous forms of cylindrical boilers were introduced. But they were superseded by the egg-end boiler. This boiler, in turn, though it is yet much used at ironworks to utilize the waste heat from the furnaces, is being superseded, for the supply of steam on a large scale, by the Cornish boiler, the Lancashire boiler, and the Galloway boiler; and of these, as before remarked, the Lancashire may be accepted as the standard type for practice on land, or stationary boilers. These boilers consist essentially of a cylindrical shell, with internal through furnace-tubes. The Cornish boiler, introduced by Trevithick in Cornwall early in the present century, has one cylindrical tube; the Lancashire boiler, having two furnace-tubes or flues, was introduced in 1844 by Fairbairn & Hetherington in Manchester; and the Galloway boiler, having two furnace-tubes merged into one of oval section, with transverse water-tubes, was subsequently introduced.

The elements of good current practice in Lancashire boilers, are now generally recognized. Mr. Lavington E. Fletcher did good service, in 1876, in giving an account of the construction, equipment, and setting of Lancashire boilers, found to give the most satisfactory results of those under the inspection of the Manchester Steam Users' Association.[1] Standard practice is exemplified in the boilers constructed by Messrs. Daniel Adamson & Co., illustrated by Plate II.

The diameter of the shell of the Lancashire boiler is suited to the size of the furnace-tube, which should be at least 2 feet 9 inches in diameter, for a suitable body of fire and convenience in stoking. Allowing a clear space of 5 inches between the two tubes, and 4 inches between the tubes and the shell at each side, a shell 7 feet in diameter is required. It affords a headway of about 2 feet 9 inches from the crown of the furnaces to the crown of the shell. A 7½-feet shell admits of furnaces 3 feet in width; but for the higher pressures that have come into practice, a 7-feet shell, with two flues of 2 feet 9 inches, is preferred.

For the length of the boiler, it varies according to taste or preference, from 21 feet to 30 feet. A length of 27 feet or 27½ feet is generally preferred. The longer the boiler the greater the liability to strain the ends by unequal expansion. The ends should be rendered elastic so as to endure the buckling action without fatigue; and for this purpose there should be not only a sufficient width of end-plate between the two furnace-tubes, as well as between them and the shell, as already explained, but also a space of 10 inches between the centre of the bottom of rivet in the gussets, and those at the furnace mouth. Five gusset-stays are inserted to bind each end-plate of the boiler to the shell. Five is a convenient number: one gusset falls on the vertical centre-line, where the front end-plate is weakest, and where it can be held fast without resisting the movements of the furnace-tubes. The part of the end-plate that should be left free is that immediately over the furnace crowns.

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[1] See Mr. Fletcher's paper "On the Lancashire Boiler," in the *Minutes of Proceedings of the Institution of Mechanical Engineers*, May 1876. To this paper the author is indebted for many useful observations.

Longitudinal stay-bolts are frequently introduced to assist the end-plates. They are placed as much as 14 inches above the level of the furnace crowns, and as close together as may be convenient. They are secured at each end with double nuts, one inside the boiler and one outside. When placed directly over the crowns of the furnaces, and only a few inches above them, they confine the tubes too strictly, and overstraining ensues.

To increase the elasticity of the front end-plate, by providing a larger free area, it is attached to the shell by an external angle-iron, rather than by flanging or by an internal angle-iron. It is not necessary to attach the end-plate at the back of the boiler, with an external angle-iron, as sufficient provision for free action is made by reducing the diameter of the furnace-tubes at the back end, so leaving more free area of plate for expansion than if the tube were carried of undiminished diameter to the end-plate. Besides, an external angle-iron fastening would be exposed injuriously to the action of the flame. Each of the end-plates of the shell is constructed of two plates welded together at the seam. By welding instead of riveting at the seam, the advantage of a flat surface is secured, convenient for the attachment of mountings. The plates are turned in the lathe at the outer edge and are bored out at the openings for the furnace-tubes.

The cylindrical shell is composed of plates about 3 feet wide between the pitch-lines of the rivets; they are laid in not more than three lengths round the circumference, in order that the longitudinal seams may clear the brickwork seatings. These seams are so arranged as to break joint, and also to avoid the centre-line along the top and the bottom of the boiler. In all the longitudinal rents obtained under Mr. Fletcher's experimental hydraulic tests, the plates bulged outwards at the middle of their width, and this action was observed to a slight extent before rupture, showing that the greatest strain, and thus the point of first fracture, occurred at or near the centre-line of each plate; and indicating that breaking joint is of practical advantage, and that a boiler composed of wide plates is not so strong as one of narrow plates.

The shells of boilers of wrought iron, 7 feet in diameter, are 7/16 inch thick for a pressure of 75 lbs. per square inch, and 9/16 inch thick for a pressure of 100 lbs. per square inch. The longitudinal seams are double-riveted, with ¾-inch rivets pitched about 2½ inches longitudinally and 2 inches diagonally. The transverse seams are single-riveted, at 2 inches of pitch. This practice is in accordance with the fact, already noted, page 621, that the stress on the transverse seams of a cylindrical boiler is only one-half of that on the longitudinal seams. Besides, by the introduction of the furnace-tubes, the area of the end upon which the steam acts is reduced, and the longitudinal tension lessened; at the same time that the ends are stayed by the tubes. End-plates are occasionally made ¾ inch in thickness for a pressure of 60 lbs. per square inch. Thicknesses of ½ inch for 75 lbs. pressure, and 9/16 inch for higher pressures, are sufficient. They are preferable to a thickness of ¾ inch, as the thinner plates yield more freely under unequal expansion. For furnace-tubes 2 feet 9 inches in diameter, $^{3}/_{8}$ inch of thickness is sufficient for a pressure of 75 lbs. per square inch, $^{13}/_{32}$ inch for 80 lbs. or 90 lbs., and $^{7}/_{16}$ inch for 100 lbs. pressure. At these thicknesses sufficient metal is provided for flanging the seams when this is required to be done; and a margin for waste by corrosion is afforded. Half-inch plates are occasionally used for furnace-tubes; but they lead to violent straining and frequent leakage at the furnace mouths and transverse seams of rivets. Plates $^{5}/_{16}$ inch thick, suitably fortified, in 2 feet 9 inch flues have been proved capable of effectively resisting a hydraulic test pressure of 150 lbs. per square inch, and of working for years under a steam pressure of 60 lbs.

Although wrought iron has been almost universally the material of steam boilers, of recent years steel of a mild quality has been extensively and successfully used; and is now almost exclusively used in the best class of work. It is stronger than iron, having a tensile strength of from 26 tons to 30 tons per square inch, against from 20 tons to 22 tons for iron plates. Ductility, as well as simple tensile strength, is required in the material of a boiler; hence the necessity for a mild quality of steel. Lowmoor iron rivets are used for iron-plate work, and they have also been used for steel-plate work. But it is now recognized that rivet steel of trustworthy quality can be produced, and steel-plating is now riveted with steel rivets. Steel, as a material for steam boilers, has occasionally failed,—inexplicably, as it appeared;—but the fitting method of treatment is now thoroughly understood, and steel of uniform quality in boilers can be ensured. There are manufacturers of boilers who use steel exclusively in their construction. The longitudinal seams of the shells are double-riveted with double welts.

The resistance of cylindrical flue-tubes to collapse under external pressure has already been discussed. It has been shown to be much less than the resistance to internal or bursting pressure. The strengthening or reinforcement of flue-tubes of Lancashire, Cornish, and like boilers, is therefore, in presence of the modern high pressures, now universally practised; so that flues may be rendered capable of resisting even higher pressures than shells, as has been repeatedly proved by experimental bursting tests, in which the shell has been burst, whilst the furnace-tubes have not suffered at all. Mr. Daniel Adamson introduced the flange-seam for furnace flues, fig. 266, in 1851,—a system now in common practice,—according to which the transverse edges of the plates are flanged outwards, and riveted together, with a ring between them. Stiffness to resist collapsing pressure is thus combined with a certain degree of longitudinal elasticity, which is provided by the bends of the plates at the flanges. The rivet-heads are in water, and there is but a single thickness of plate exposed to the direct action of the fire. For the strengthening of the furnace-flues of old boilers, made with ordinary riveted seams, the simplest appliances are hoops of angle-iron, as in figs. 267, encircling the tube, with an interspace

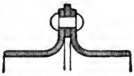
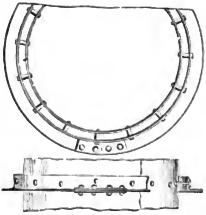


Fig. 266.
Adamson's Flange-joint for Furnace-tubes.

or clearance for circulation from 1¼ inches to 1½ inches wide, between the hoops and the tube. They are riveted together, with ferules in the interspace. Double angle-irons, or angle-irons in pairs, are occasionally employed. But these, of course, increase the width of interspace between the fire and the water, and incur liability to overheating. On another plan, fig. 268, the furnace-tube is encircled by a few thin rings of iron on edge, 3 inches or 4 inches in width, and ⅝ inch thick, closely fitted to the tube, but not attached to it. This plan is obviously imperfect. Rings of **T** iron forming the connection between the plates at the circular seam, fig. 269, were early introduced, it is believed, by Messrs. Hick of Bolton, in conjunction with welded longitudinal seams. This system has been widely employed for strengthening furnace-tubes. The rings form a strong support so long as they continue in good order. But, whilst the horizontal member is exposed to the heat of the furnace, the vertical member, or rib, is surrounded by water, and is comparatively cooler than the riveted section. The consequence is unequal rates of expansion, and an undue straining and radial fracturing of the rib. A fracture, once commenced at the circumference, gradually advances inwards, and is at the same time widened by corrosion. Such a liability is objectionable, and even dangerous. The Bowling hoop, fig. 270, was introduced by Mr. Thomas Hill in 1861, when it was made from railway bars rolled at Bowling. The Bowling Iron Company manufacture such hoops in one piece of steel. The Bolton Steel and Iron Company manufacture a ring of like section.


Figs. 267.—Angle-iron Hoop for strengthening Boiler Flue-tubes.

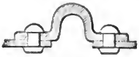


Fig. 270.—The Bowling Hoop for Furnace-tubes.

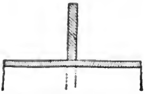


Fig. 268.—Strengthening Ring for Furnace-tubes.

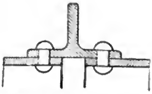


Fig. 269.—T-iron Joint for Furnace-tubes.

Messrs. Hawkesley, Wild & Co. manufacture furnace-tubes, consisting of rings of alternating larger and smaller diameters, subsequently illustrated, the smaller rings being expanded at their ends to join the larger rings by a seam of rivets. The resistance to collapsing pressure thus gained is evident, whilst the tube is free to expand radially as well as longitudinally. The smaller rings of the two furnace-tubes applied to a Lancashire boiler are opposed to each other, and so provide at such intervals convenient room for a man to stand between the tubes, for purposes of inspection, &c. The additional room, too, is useful no doubt in facilitating circulation of water; whilst the varying section aids in promoting the mixing and the combustion of the gases. See illustrations of the boiler, pages 717 and 718.

Mr. J. N. Paxman has obtained good action from the joint formed by setting out the edges of the plates of the furnace-tube, and forming the lap-joint on a larger circle than that of the body of the tube, as in fig. 271. It greatly increases the power of resistance of the furnace-tube to collapse; whilst it acts as an expansion-joint, and the rivets at the junction are out of the range of the scouring action of the flame. The flues of boilers with this joint are made of solid welded rings in steel or iron, and rolled out accurately with a machine to the form shown in fig. 272.

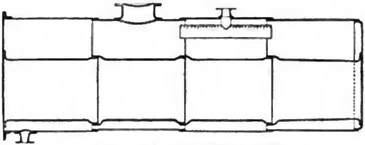


Fig. 271.—The Paxman Furnace-tube.

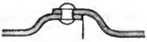


Fig. 272.—The Paxman Joint for Furnace-tubes.

Mr. S. Boswell has designed a flange-seam, fig. 273, of a conical form, for which each plate is welded into a ring, of the form of a frustum of a cone. The seam yields to expansion and contraction in any direction, and each ring being conical, the heated gases, as they advance, are disposed to bear or press upon the narrowing flue; with an accelerated transmission of heat in consequence. Suppose, for instance, that the smaller diameter is 4 inches less than the larger, assuming the latter to be 3 feet. The respective extreme sectional areas are as 5 to 4, approximately, and the velocity is accelerated in the inverse ratio, as 4 to 5. It is probable that the eddies consequent on such alternations of velocity not only augment the transmission of heat, but also promote the prevention of smoke.

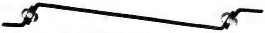
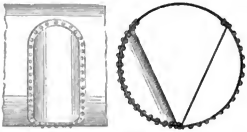


Fig. 273.—Boswell Flange-seam.

In some cases, Petrie's water-pockets, figs. 274, and in others, Galloway tubes, fig. 275 and Plate II., are introduced into furnace-tubes as a precaution against collapse, in addition to their acting as promoters of circulation. The Galloway tube is taper, to admit of the lower flange being



Figs. 274.—Petrie's Water-pockets.

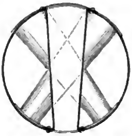


Fig. 275.—Galloway Tubes.

passed through the upper opening, to get the tube into its place. Many makers insert cylindrical tubes or pipes and weld them in.

The most notable advance in furnace-tubes, of recent years, is the corrugated furnace-tube introduced by Mr. Samson Fox some years ago, illustrated by fig. 276, and in Section IV. They are manufactured of mild steel at the Leeds Forge. As an illustration of the comparative strength of corrugated and plain flue-tubes, two flues of like material and dimensions, one corrugated and the other plain, were crushed by hydraulic pressure, when the corrugated flue sustained a pressure of 1020 lbs. per square inch before giving way, whilst the plain flue failed at 225 lbs.

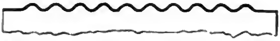


Fig. 276.—Fox's Corrugated Furnace-tube; Leeds Forge.

A modification of the corrugated flue, specially adapted for stationary boilers, has been introduced by Messrs. Hopkinson & Co., consisting of portions of the flues, at regular intervals or where desired, being left uncorrugated for the introduction of Galloway tubes, the plain parts being of the same diameter as the tops of the corrugations.

As stays in supporting the end-plates to which furnace-tubes are attached, the Leeds corrugated flues are inferior to plain flues. They elongate under pressure; and when they chance to be overheated, they become still less able to resist the forces tending to elongate them, and they may be drawn out till an excessive strain is thrown upon other parts. This contingency may be counteracted by the insertion of longitudinal stay-bolts around the tube.

The Farnley Iron Company manufacture boiler-flues of mild steel, with "spiral" or helical corrugations, as in fig. 277. They tend to shorten under pressure. In one case, under a pressure of 200 lbs. per square inch, 3½-feet flues, 7¼ feet long, shortened $^1/_{32}$ inch, resuming their normal length on being relieved. Thus they serve as stays for the ends of the boiler. The direction of the corrugations facilitates the draught, presenting an easy angle for impingement. The collapsing pressures for flues of 3½ feet outside diameter were proved to be as follows:—

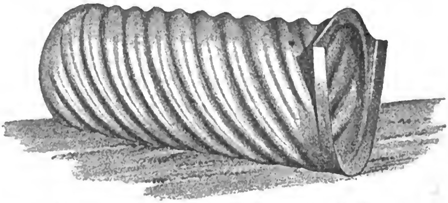


Fig. 277.—Helically-corrugated Furnace-tube. Farnley Iron Company.

9/16 inch thickness......	847	lbs. collapsing pressure per square inch.
7/16 " "	635	" " " "
5/16 " "	526	" " " "

Short stand-pipes of wrought iron, for the attachment of fittings—called "fitting-blocks" when made, as formerly, of cast iron—are riveted to the shell of the boiler, and strengthen the plate round the hole. The manhole mouthpiece, also, is made of wrought iron: it was first adopted by Mr. Daniel Adamson. The results of Mr. Fletcher's bursting tests show that cast-iron pieces in such situations are relatively weak and ought to be abandoned for wrought-iron work.

The form of the Lancashire boiler remains unaltered when the boiler is filled with water. But there is a very sensible alteration of form by the action of the fire. The tubes hog upwards in consequence of unequal heating, and thrust out the ends. This action is most severe soon after the fires are lighted. But when the boilers become generally heated, the distortion ceases. It seldom lasts more than an hour. Mr. Fletcher records in connection with the tests conducted at Wigan, already noticed, page 144, &c., the results of his observations on such distortion by means of gauges, summarized in the following table, No. 151.

Table No. 151.—DISTORTION OF FURNACE-TUBES IN INTERNALLY-FIRED BOILERS, FROM THE ACTION OF THE FIRES. (Wigan Trials.)

Lancashire Double-flue Boilers of Iron and of Steel.

CONDITIONS (Averages of several Observations).	Positions of Gauges on Furnace-tubes.		
	One-fourth from Front End.	Mid Length.	One-fourth from Back End.
	inch.	inch.	inch.
1. Rise of tubes in boilers; flame passing through external brick flues...............	3/8	3/8	5/16
2. Rise of tubes in iron boiler; flame passing direct to chimney	1/2	7/16	3/8

The boilers referred to in this table were 28 feet in length and 7 feet in diameter, with two furnace-tubes, 2 feet 7½ inches in diameter. The shell-plates of the steel boiler were fully 3/8 inch in thickness, and those of the iron boiler 7/16 inch. The tubes were respectively 5/16 inch and 3/8 inch thick. Care was taken not to strain the boilers by severe firing, steam being got up with the dampers only partially open.

In the second case, of the boiler treated as for a flash-flue, when the draught was stopped off from the side and bottom flues, the distortion was greater than in the first case. The shells when raised to 212° F. lengthened 1/8 inch in 8 feet. Of course, when expansion proceeds unequally above and below, hogging of the shell results. It is shown by the table, too, that the rise of the tubes was greater near the front than near the back. When hard pressed with a heavy fire, in regular action, furnace-tubes with flanged joints may give way at the bottom of the tubes; and the straining and leakage may be more severe in proportion as the end-plates are more rigid, and the longitudinal stay-bolts are tighter, and brought down nearer to the angle-iron connections of the tubes to the end-plates. This result is brought about by the application of the fire to the upper side of the furnace-tube, elongating the upper surface, causing the tube to arch upwards, and opening or easing the flange-joints at the caulking, which is inside. Mr. Fletcher mentions a case in which the furnace-tubes had leaked repeatedly at the bottom. The seams were caulked and re-caulked without success, until at length the nuts at the ends of the longitudinal stay-bolts were let out. The stay-bolts were thus partially released, and the leakage was cured.

The foregoing observations lead to a consideration of the familiar form of failure known as grooving:—the result of overstraining transversely—excessive bending strain. In Lancashire boilers the grooving action takes effect chiefly in the front plates, inside, just over and around the furnace mouths. They are the joint results of mechanical action and chemical action. The plate is alternately forced or bent outwards by the movement of the furnace-tubes, and restored to its normal position, corresponding with the intermittent action of the fire, until it opens ever so slightly, forming a slit which is at first scarcely visible, and which gradually extends into the plate. The slit or crack is penetrated by the water, and if the water be acidulous, it gradually corrodes the metal in the slit, and widens the slit so as to form a groove, where the plate is strained between two angle-irons—the angle-iron joining the furnace-tube to the end-plate, and the angle-iron of a gusset-stay. The angle-iron, joining the furnace-tube to the end-plate, is also subject to grooving. When four gussets, instead of five, the usual number, connect the end-plate with the barrel of the boiler, the outer gussets and their angle-irons are unfavourably close to the tube angle-irons. The gussets act as powerful ties to prevent the end-plate from yielding to the thrust of the heated tube. Hence the overfatigue and consequent grooving of the intermediate metals—the plate and the angle-iron.

The grooving of the T form of strengthening rings applied to furnace-tubes has already been noticed; and there is a third form of grooving, which takes place next the longitudinal lap-joint seams of cylindrical boilers. The author publicly directed attention to the causes of this kind of grooving in 1860.[1] "The alternate tension and relaxation of the plates at the joints, as the steam is got up and let down, are attended by an alternate distortion, incipient, it may be, and resumption of the normal form, a bending and unbending of the plates on each side of the joint; in consequence of which the texture of the metal is gradually loosened, in lines near to, and parallel to the joint, and it is thus laid open to corrosive action. On this interpretation, the commencement of a groove or furrow, establishing a weak place, and concentrating the action there, would suffice to extend and deepen it, to the dangerous limits occasionally announced by explosions.

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[1] *Recent Practice in the Locomotive*, 1860, page 15. See also *Railway Locomotives*, 1860; by D. K. Clark.

"The weakness attendant on lap-joints is strikingly exemplified in the lap-welded joint, when subjected to extreme tension. The tensile strength, though the metal at the weld may be perfectly solid and fully as strong in itself as the body of the plate, is much below that due to the regular section of the plate. Here there is no sign of elementary weakness due to the reduction of metal by rivet-holes. The inferiority of strength arises solely from the bending of the plates on both sides of the lap, and the over-straining of the boiler, in the endeavour to attain to the position of stability."

"That the furrowing of plates at the riveted joints results from the indirectness of the strain of the steam pressure, is rendered still more probable by the analogous furrowing which results from reciprocating strains of another kind. In the more ancient classes of locomotive engines, in which the cylinders are fixed to and work from the smokebox plates, the alternate forward and backward strains by the steam pressure on the piston, have been observed to weaken and to subject to corrosion and leakage, the substance of the plate along the edge of the angle-iron at the junction with the barrel."

The producing cause of grooving of longitudinal cylindrical rivet-seams, as well as of end-plates here pointed out, is now universally accepted in explanation of the grooving or furrowing of boiler-plates.

The author likewise, in 1860, pointed out for the first time, the advantage of making the longitudinal joints of cylindrical boilers stronger than the transverse joints, for the reason that the proportional stress on the former was twice as great as on the latter.[1] "The longitudinal seams, therefore," he says, "should be the strongest, and there, setting aside meantime the welded joint, the double-welt double-riveted joint, should be applied; and for the circular or transverse seams, bearing only half the strain to which the longitudinal seams are subjected, the single-riveted lap-joint is sufficient, being, in fact, proportionally stronger in its place than the other:—proportionally stronger, that is to say, with respect to strains arising from steam pressure; and when, as it ought to be, the boiler is employed exclusively as a generator of steam." . . . "A boiler riveted by double-welt double-riveted longitudinal joints in the cylindrical portions, and everywhere else by single-rivet lap-joints, is practically of equal strength throughout."

Since the foregoing was published, it has become the practice, nearly universal, to double-rivet the longitudinal seams of cylindrical boilers, either with lap-joints or double-welt joints, and to single-rivet the circular seams. But the circular seams are sometimes double-riveted with lap-joints or welted joints.

In the construction of Lancashire and Cornish boilers, the plating of the shell is truly cylindrical; that is to say, having inner and outer belts alternately, the laps of circular seams directed alternately towards the front and the back. But for externally-fired boilers, "thimble-plating" is adopted; that is, conically-formed belts, one to enter the other, so that all the laps are turned from the fire.

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[1] *Recent Practice in the Locomotive*, pages 11, 12. See also *Railway Locomotives*.

Steam-domes were formerly applied to stationary boilers, as in fig. 278, to augment the steam-space and provide dry steam. That they were effective for this purpose need not be doubted; but they, of course, destroyed the balance of pressure, and were a source of weakness. Mr. E. A. Cowper, in order to compensate for the want of balance, applied a straight horizontal stay-plate across the boiler, underneath the base of the dome, as a tie from side to side, so as to make up the resisting power of the boiler at that place. This plate is perforated; so also is the shell of the boiler under the dome, the shell not being cut away. In other cases, the hole opening into the dome from the boiler is only made the usual size of a manhole.

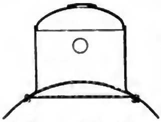


Fig. 278.—Steam-dome, by E. A. Cowper.

Steam-domes are not now applied. They have been superseded by an internal perforated pipe, placed along the top of the boiler longitudinally, first employed in stationary boilers by Hick. The volume of the dome, of course, adds but little proportionally to the steam room of the boiler; but it carries the steam to a high level above the water, and in this way priming water is dropped behind. The perforated pipe answers the purpose very well. It operates on a different principle from that of the dome,—collecting the steam at many points distributed over a considerable extent of the steam-space, and so obviating undue concentration of the upward streams of steam, and the consequent uplifting or priming of water. The receiving pipe is of cast iron, about 5 inches in diameter and 6 feet in length, closed at the ends, and suspended from the crown of the shell by a stud and set-screw at each end. There are, say, 100 rectangular perforations, usually about 1½ inches long by ¼ inch wide, in the upper part, in three rows, through which steam enters the pipe; and the steam leaves the pipe midway through a nozzle, say, 4¾ inches in diameter, which fits a 6-inch opening through the shell into the stand-pipe of the steam stop-valve.

Where a reservoir for steam is required, a separate vessel should be provided, connected by one or more necks to the boiler, limited in diameter, so as not materially to weaken the shell by the openings. The weakness of the shell where it is cut out for a dome has been proved by actual test.

The manhole is made with a mouthpiece of wrought iron, welded into one piece, flanged at the top and bottom, and double-riveted to the boiler, say, 16 inches in diameter, of ¾-inch plate. A binding hoop is added below the shell-plate, and is riveted into the structure. The resisting strength of this mouthpiece has already been noticed, page 682.

The mudhole at the front of the boiler, beneath the furnace-tubes, is fitted with a substantial mouthpiece, fixed on the inner side of the end-plate.

Fittings.—The fittings are so arranged that those to which frequent access is required are immediately within reach of the attendant when standing in front of the boiler. The feed-water is introduced on one side of the front end-plate, about 4 inches above the level of the furnace crowns, an internal pipe being carried along inside the boiler for a length of, say, 12 feet, and perforated for the last 4 feet of its length, for the purpose of at once dispersing and diffusing the cold feed-water introduced. On the other side of the front end-plate, the scum tap is fixed, for the purpose of drawing off from time to time the froth or scum that floats on the surface of the water, which is collected in a series of sediment-catching troughs fixed inside the boiler. Two glass water-gauges are fixed to the middle of the end-plate, a pointer showing the correct height at which the water should be maintained. A dial pressure-gauge is mounted immediately above the water-gauges; and above all, on the top of the boiler, a dead-weight safety-valve. A low-water safety-valve is placed on the middle of the boiler. Below, underfoot, is the blow-out tap. A handle for regulating the dampers is frequently placed at the front of the boilers.

The position of the feed-water inlet-pipe is a point of importance. If the feed-water be cold and be introduced near the bottom of the boiler, it is likely to induce local contraction, and thereby strain the transverse beams of rivets at the bottom of the shell. But when introduced near the surface of the water and passed through an internal perforated pipe, it is dispersed before falling to the bottom, and acquires in a greater or less degree the temperature of the surrounding water. A non-return or check valve is, in many cases, applied in the feed-pipe when the feed is delivered at a low level; but it sometimes fails and allows the water to escape from the boiler, acted on by the pressure of the steam. The crowns of the furnaces in this case become bare and overheated, and the boiler may even be emptied. But when the feed is delivered at a high level, as already described, the boiler cannot be drained to any dangerous extent by leaky valves.

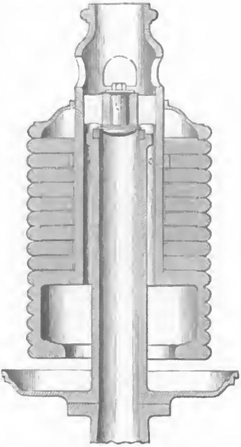


Fig. 279.—Dead-weight Safety-valve.

The dead-weight safety-valve, which is of the Cowburn type, shown in section, fig. 279, is simple and efficient. The centre of gravity of the load being below the seating, the valve remains naturally in position, without the aid of wings or fins, and is free from frictional resistance, or liability to stick fast. The valve is loaded with flat annular plates or rings, and the shell is cast with mouldings around it at the lower part, uniform with the weights. The material is so adjusted that each moulding, as well as each annular plate, represents a pressure of 5 lbs. per square inch on the valve. The diameter of valve generally adopted is 4 inches, which requires approximately a load of 8 cwt. for a blowing-off pressure of 70 lbs. per square inch, and 11 cwt. for 100 lbs. pressure. The great weight required to load this valve is a mark of safety in respect of overloading, since additional deadweight of a considerable magnitude would be required seriously to augment the blowing-off load.

It is of importance that the safety-valve should be placed on a mouthpiece distinct from that of the steam stop-valve, in order that the safety-valve may at all times be free to act, even when a plug may have inadvertently been left undrawn from the steam-pipe. In cases where the safety-valve has been placed on the same outlet as the steam stop-valve, the accidental plugging of the pipe has given rise to explosions: the safety-valve blowing off steam contributed from the main pipe, whilst the steam was, in fact, pent up in the boiler.

The low-water safety-valve is of the Hopkinson type. To one end of a lever inside the boiler, a float is attached, so that when the water falls below the desired level the float falls also, and thus raises the valve, and allows the steam to blow off. Attention is thus drawn to the low level, and the pressure is lowered at the same time. The valve is compound, one valve being seated on the other. The central disc is loaded by a dead-weight inside the boiler, and is acted on by the lever in the event of low water. The annular or outer valve is loaded by an external lever and weight; and lifts along with the central disc in the event of high pressure of steam. Thus it is that the valves blow off on the occurrence either of too high a pressure of steam, or too low a level of water. The outer valve is 5 inches in diameter, and the inner valve is 2½ inches. The action of the steam-valve can be tested by easing the lever when the steam is up; and the low-water apparatus can be proved by blowing off water at the blow-out tap and lowering the water-level to, say, 6 inches above the crowns of the furnaces.

The furnace mouthpieces are of wrought iron, finished off with a neat brass beading, leaving in view the circle of rivets round the tube. The fire-doors are fitted with a sliding ventilating grid on the outside and a perforated box baffle-plate on the inside, the aggregate area of air-passages being about 50 square inches for each door, or about 3 square inches per square foot of fire-grate. The fire-grate is 6 feet long, having the bars in three equal lengths, about ¾ inch thick, and spaced ⅜ inch apart, as air-space. The bearers consist of two wrought-iron bars carried on wrought-iron brackets riveted to the sides of the furnace-tubes. The standard length of grate is 6 feet. Shorter grates are more economical, inducing greater evaporative efficiency; but the concentration of the fire is more trying to the boiler, and it has been found, where the feed-water has not been good, to injure the plates of the furnace, and to render necessary the lengthening of the grate.

The bridge at the end of the fire-grate is of fire lumps, laid in a seat of cast iron. The top of the bridge is 14 inches below the crown of the furnace.

The boiler is set on side walls, and rests on fire-brick seating blocks presenting a bearing surface, say, 5 inches wide. The side flues are, say, 6 inches wide at the top; they are carried up to the level of the furnace-crowns, or a few inches above it, and down to the level of the bottom of the shell. The width of the bottom flue is equal to the radius of the boiler, and the angle of the bearing of the seating block is 30°, for any diameter of shell. The flame, after leaving the furnace-tubes, passes under the bottom of the boiler, and returns to the chimney along the side flues. According to other practice, which was common some years ago, the draught is conducted from the furnace-tubes direct to the side flues, and thence below the bottom of the shell. But, on this system, the circulation of water is but slowly promoted; so that, in getting up steam, the top of the boiler becomes hotter than the bottom, and straining at the transverse seams ensues. If, in addition to this, the feed-water, when cold, be pumped in at or near the bottom of the boiler, the straining is augmented. The occurrence of seam-rents at the bottoms of Lancashire boilers, in cases where the final traverse was made under the shell, has led to the abandonment of that plan, and the general adoption of the plan already described, of carrying the burnt gases direct from the furnace-tubes under the bottom; and also the introduction of the feed-water near the surface.

Whichever course may, in itself, be the more economical of fuel, the use of feed-water heaters, or fuel economizers, as a means of exhausting the burnt gases of a large portion of their heat on leaving the boiler, meets and solves the question in a practical way. By means of an effective economizer, the temperature of cold feed-water may be raised to from 260° to 320° F., gaining an accession of, say, 240 degrees of temperature. Besides the direct saving of fuel effected, the heating of the feed-water prevents local cooling, with undue straining, and obviates repairs.

To compare the sectional areas of flueway with the area of the grate, in ordinary practice, the throat of the furnace or passage-way over the bridge, is 14 inches high at the crown of the tube, and has a sectional area of 1.88 square feet, about one-ninth of the area of the grate. The furnace-tube beyond the bridge has a sectional area of 5.94 square feet, which is nearly one-third of the grate-area. The bottom flue is 3½ feet wide, with a depth under the boiler averaging 2 feet, making a sectional area of 7 square feet, or fully one-fifth of the grate-area. The side flues are 4 feet 7 inches high, and are 6 inches wide at the upper part, widening towards the lower part. They have a united area of 12½ square feet, or about two-fifths of the area of grate. A brick chimney adapted for the service of six boilers is, say, 11 feet in diameter at the bottom and 6½ feet at the top; and is 150 feet high. The sectional area at the bottom is 95 square feet, and at the top it is 33 square feet. These quantities compared with the total grate-area for six boilers, $(33 \times 6 =)$ 198 square feet, are respectively about one-half and one-sixth of the total area of grate.

The foregoing proportions may be tabulated thus:—

Table No. 152.—LANCASHIRE BOILERS:—SECTIONAL AREAS OF FLUEWAY.

Locality.	Area of Fire-grate.	Area of Flueway.	
	square feet.	square feet.	in parts of grate-area.
Air-space through one grate......	16½	5½	1/3, or 33 per cent.
Over one bridge....................	16½	1.88	1/8.8, or 11½ "
One furnace-tube	16½	5.94	1/2.8, or 36 "
Bottom flue..........................	33	7.00	1/4.7, or 21 "
Two side flues......................	33	12½	1/2.6, or 38 "
Chimney at base (6 boilers)......	198	95	1/2.1, or 48 "
Do. at top do.	198	33	1/6, or 17 "

The flooring or hearth-plates at the front of the boiler, covering the hearth-pit, are set so as not to butt against the boiler, but so as to be entirely below it, leaving the whole of the front end-plate open to view. They are carried on framing, and can be easily lifted. The hearth-pit is about 3 feet wide and 2½ feet deep, and it extends from side to side of the boiler-house. In it is laid the main feed-pipe, as well as the discharge pipe from the blow-out and the scum tap. The flue-doors open into this hearth-pit. The face of the brickwork at the front of the boilers is set back about 6 inches from the end of the boiler: otherwise stated, the front of the boiler is 6 inches in advance of the brickwork, leaving the angle-iron junction of the end-plate with the shell perfectly open. The brickwork beneath the boiler at the front is recessed so as to clear the blow-off elbow-pipe. The tap and pipe are thus free to move should settlement of the boiler take place.

The boiler is covered with an arch of brickwork, leaving a space of about 2 inches between it and the shell, which is filled with cork shavings or a coating of a non-conductor composition. Round the fittings a clear space is maintained, leaving the rivets of the junctions exposed. Sometimes only a coating of composition is applied.

The steam generated in two or more boilers is usually collected in a main pipe laid transversely to the boilers. Here, as elsewhere about a boiler, rigidity of connection is to be avoided. The connection of the boiler with the main steam-pipe is, for this reason, made by means of a "radial pipe," or a "springing length" of pipe between the stop-valve or nozzle and the main steam-pipe. As shown in fig. 280, page 703, it is a 6-inch pipe of cast iron, 5 feet or 6 feet long. Mr. Daniel Adamson makes the pipe of wrought iron for the higher pressures. A main pipe, serving two boilers, would be 8 inches in diameter. For a range of, say, six boilers, a 12-inch main is provided.

If the main pipe were bolted direct to the stop-valve, the joints would be strained by the rising and falling of the boilers when steam is got up in them, or when they laid off. When the main pipe is of considerable length between the boilers and the engine, it should not be taken in a direct line, but should follow a course with one or two bends, to give elasticity for the incidental alternating movements by expansion and contraction. Long lengths of pipe have been, in some cases, divided by expansion-diaphragms, which consist of two circular discs or sheets of wrought iron, riveted together with a distance ring between them at the periphery, and connected to the pipe at the centre, one disc to each pipe-end. The discs are cut out to form a central opening of the diameter of the pipes, affording an uninterrupted thoroughfare for the steam, which, at the same time, has access to and fills the interior of the diaphragm. It is obvious that a longitudinal bulging action is set up, counteracting the expansion and thrust of the pipes; and Mr. Fletcher mentions a case in which the main junction-valve was broken off by the thrust occasioned by the bulging of the expansion-diaphragm.

The connecting length of pipe between the stop-valve and the main pipe is connected horizontally to the main pipe at the side, as shown in fig. 280, page 703. Such connections are sometimes made to incline upwards, so that the water may drain back to the boilers. This plan is objectionable; as condensed steam is accumulated in the pipe when the boiler is laid off, and interferes with the steam from the boiler when set to work again. The water is liable to be carried forward by the rush of steam, giving rise to water-hammer, and tending to burst the pipes. Mr. Fletcher mentions an instance in which the steam stop-valve was fractured by the reaction resulting from the condensation of steam and the formation of a vacuum in the steam-pipe which followed on shutting off the steam shortly after it had been turned on, and had begun to force its way through a body of water in the pipe. The steam-pipe should be laid to drain towards the engine, the water being intercepted by a trap or a separator.

The weight of a Lancashire boiler such as that which has been described, 7 feet in diameter and 27 feet long, made of 7/16-inch plates, is about 12 tons without fittings; fittings weigh 3½ tons, and make a total of 15½ tons—something over half a ton per lineal foot. The cost is about £425, being at the rate of about £27, 8*s*. per ton, or £16 per lineal foot.

The heating surface of this boiler is as follows:—

	Square feet.
Shell	370
Furnace-tubes, without water-pipes	450
Water-pipes	30
Total heating surface	850

The total area of the fire-grates, each 6 feet long and 2 feet 9 inches wide, is 33 square feet. The ratio of grate-area to heating surface is as 1 to 26.

In feed-water heaters, from sixty to sixty-four pipes of the Green Economizer class, are frequently introduced per boiler, each having 10 square feet of heating surface, measured externally, making a total surface of 600 square feet, which is nearly three-fourths of that in the boiler. But a much less number is frequently met with. Mr. Adamson places only forty-three pipes per boiler for the half-dozen boilers shown in Plate II. But, of course, if boilers out of action be excluded, the number of pipes per boiler would be greater.

The quantity of coal that may be consumed in this boiler without distressing it, is from 15 tons to 20 tons per week of 60 working hours, or from 5 cwt. to 6½ cwt. per hour, or 17 lbs. to 23 lbs. per square foot of fire-grate per hour.

CHAPTER X.—LANCASHIRE STEAM BOILERS (*continued*).

LANCASHIRE STEAM BOILERS BY DANIEL ADAMSON & CO.

A range of six Lancashire boilers, as constructed by Messrs. Daniel Adamson & Co., Dukinfield, is represented in fig. 280 and in Plate II., with brick setting, flues, and economizer. The mountings comprise the following items:—

Dead-weight safety-valves; high-steam and low-water safety-valve; steam-nozzle; anti-priming pipe; round manhole; oval manhole; check feed-valve; perforated feed-pipe; brass scum tap; scum troughs; brass blow-off tap; brass water-gauges; brass water-level pointer; steam indicator; wrought-iron fire-fronts; fire-grate; flue-doors; floor-plates; circulating tubes; slide dampers; swivel dampers.

The course of the draught is shown by the arrows. The burnt gases, after having passed through the furnace-tubes, descend and return under the bottom of the boilers to the front, where they divide and repass along the side flues to the back. The boilers are 30 feet in length and 7 feet in diameter, each having two flues 2 feet 9 inches in diameter. Each flue is crossed by five circulating water-pipes, conically formed, being wider at the upper end than at the lower end. The shell and the furnace-tubes are constructed of ten rings of plates about 3 feet wide between the centre-lines of the rivet-holes. The boiler is constructed either of iron or of steel, of the best material: the iron plates used having a tensile resistance of from 24 tons to 25 tons, elongating from 10 per cent to 15 per cent before fracture; and the steel plates, a tensile resistance of from 27 tons to 32 tons, with an elongation of about 30 per cent. Great numbers of boilers have been and are now made of steel by Messrs. Adamson & Co., dating from the year 1858, and having worked and being still at work quite satisfactorily. For pressures of 70 lbs. per square inch and upwards, they seldom make any but steel-shell boilers. The plates are planed at the edges, and the rivet-holes for the joints are drilled through the two plates together, after the plates have been exactly formed for the positions they are to occupy. Thus sound mechanical work is ensured,—the holes perfectly matched, the plates perfectly fitted, and distressing of plates and indirectness of rivets obviated. The furnace-tubes are solid-welded into rings at the longitudinal seams, and the water-pipes are welded solid into the flue-rings. The circular joints are put together with Mr. Adamson's anti-collapsive flange-seam,

which has already been described, fig. 266, page 689. The flanges formed on the rings are turned at the edges. By this system of construction, neither rivets nor laps or double thicknesses of plate, are directly exposed to the action of the fire.

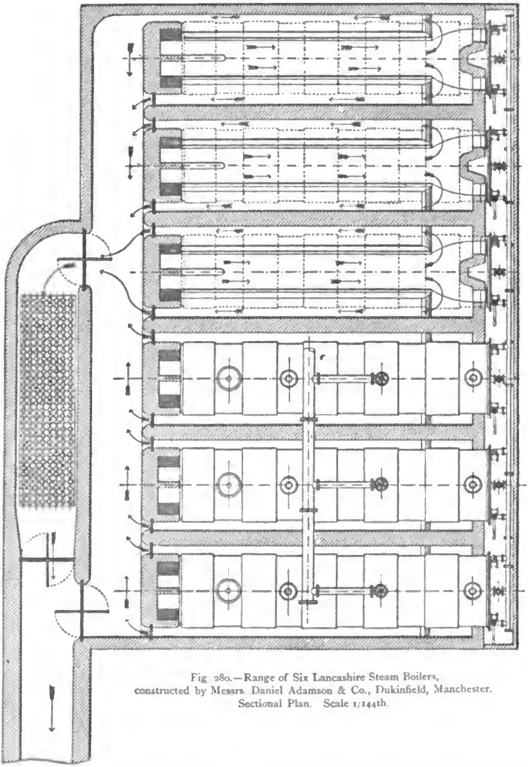


Fig 280.—Range of Six Lancashire Steam Boilers,
constructed by Messrs Daniel Adamson & Co., Dukinfield, Manchester.
Sectional Plan. Scale 1/144th.

The boilers shown here and in Plate II. are constructed for a daily working pressure of 80 lbs. per square inch. For a boiler of steel, ⅜-inch plates are used; of iron, ½-inch plates. The longitudinal seams of the shell are double-riveted; the circular seams are single-riveted. But for pressures higher than 80 lbs. per square inch, the circular seams are double-riveted; and for the highest pressures, the seams of the shell are butt-jointed, with four rows of rivets.

The fire-grates are each 6 feet in length, and are laid in three lengths of bars, which are of cast iron, 2 feet each, making a total area of 33 square feet of fire-grate for each boiler. The bridge is of fire-lumps laid on a cast-iron seating fixed in the flue. The top of the bridge is about 12 inches below the crown of the furnace, against 14 inches in the typical boiler already described. The system of flues and the course of the draught, are the same as already described for the typical boiler. But the side flues are 12 inches wide at the upper part, against 6 inches in the typical boiler; whilst the bottom flue is 3 feet deep against 2 feet. The back flue, or collecting flue, at the back of the range of boilers, into which the side flues of the boilers discharge their currents, is 5 feet wide, and about 9½ feet high; and it discharges into the main flue to the chimney, of the same dimensions. It is backed by an alternative flue, containing a Green's economizer, made 6 feet in width, and 11½ feet high, to give room for the economizer. When the economizer is not in action, the damper opening to it is closed, and the draught is conducted direct from the back flue into the main flue to the chimney. When the economizer is in action, the back flue is shut off by a damper from the main flue, and the draught is turned into the economizer flue. With such roomy flues, the force of draught from the several boilers in the range is nearly uniform. The chimney designed for these six boilers is 11 feet in diameter at the base, 6½ feet at the top, and 150 feet high.

The economizer pipes, 256 in number, are 9 feet long. They present an area of 2764 square feet of heating surface, calculated externally, or 461 square feet per boiler.

The heating surface of each boiler is 856 square feet.

The nominal power of each boiler is said to be 60 horse-power. This nominal value is calculated from the combined horizontal area of the shell and the two flues in square feet, dividing it by 6,—allowing about 6 square feet of horizontal area per nominal horse-power. In this instance, the sum of the diameters of the shell and the flues is $(7+(2\frac{3}{4}\times 2)=)$ 12½ feet, and $(12\frac{1}{2}\times 30=)$ 375 square feet is the combined horizontal area. Then $(375\div 6=)$ 62½, the nominal horse-power precisely. The grate-area 33 square feet, is at the rate of a little more than half a square foot per nominal horse-power; and the heating surface, 856 square feet, is a little over 14 square feet per nominal horse-power.

The fuel consumed is from 14 lbs. to 17 lbs. per square foot of fire-grate per hour, or from 4 cwts. to 5 cwts. total per hour.

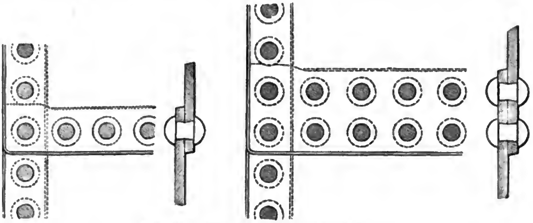
The weight of each boiler, in steel, with mountings complete, is from 16 tons to 17 tons. The price is about £390, or from £23 to £24 per ton. The usual cost of setting a single boiler is about £70. A number of boilers are set at a less rate per boiler.

LIVET'S SYSTEM OF BOILER SETTING.

The system of boiler setting introduced by Mr. Fountain Livet, has already been described and illustrated at page 140, as applied to a single steam boiler at the works of Messrs. Clay & Sons, London. As applied to a range of Lancashire boilers at the Thames Sugar Refinery, Silvertown, belonging to Messrs. Henry Tate & Sons, results of its performance have been given, page 198. The boilers, four in number, are of the largest size, being 7½ feet in diameter, and 34 feet in length. The flues from the boilers are entirely independent of each other, and are joined separately to the chimney. The chimney is 60 feet high above the ground-level, and is itself divided vertically into four compartments, one of which is devoted to each boiler. By this arrangement the flueway and draught for each boiler are entirely separated from those of its neighbours, from the furnace to the top of the chimney; and the chances of counteraction are avoided. The chimney is parallel-sided externally; whilst the internal vents are wider at the top than at the bottom, following out Mr. Livet's principle of progressive enlargement of flueway from the fireplace to the top of the chimney.

CHAPTER XI.—THE GALLOWAY STEAM BOILER.

The Galloway boiler, Plate III., manufactured by Messrs. Galloways, Manchester, resembles the double-flue Lancashire boiler externally, with two furnace tubes; but, internally, it is distinguished by the wide segmental, kidney-shaped flue-tube, into which the furnace-tubes are merged, and which reaches from the point of junction of the tubes to the far end of the boiler, and is traversed by closely set water-tubes or pipes. The boiler is made of Snedshill iron plates, the ultimate tensile strength of which is from 19 tons to 22 tons per square inch, longitudinally, or with the fibre; and from 18 tons to 21 tons per square inch, transversely. The shell is 24 feet long, and 7 feet in diameter, made of eight rings of plates, 7/16 inch thick, in 3-feet lengths, between the centre-lines of the rivet-seams.



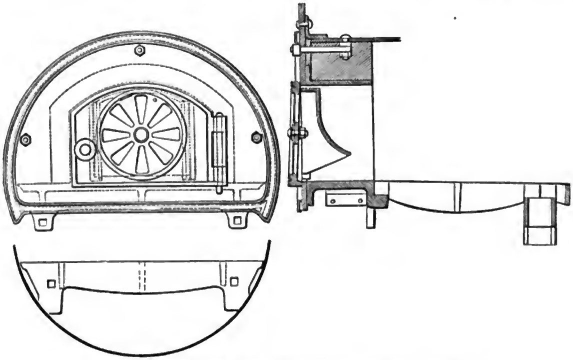
Figs. 281.—Riveted Seams of the Galloway Boiler.

The longitudinal seams are double-riveted, or chain-riveted, breaking joint, and the transverse seams are single-riveted as in figs. 281. For the single-riveting, the lap is 2¼ inches, and the rivets are ¾ inch at a pitch of 2 inches. For chain-riveting, the lap is 4¼ inches, and the pitch is 2¼ inches. The edges of all the plates are planed and fullered, not caulked. The rivet-holes are drilled after the plates are put together. The ends of the boiler are each in one plate, ½ inch thick, turned at the edge, and attached to the shell at the front by means of a solid welded ring of angle-iron 3 inches by ½ inch in average thickness, applied on the outside of the shell. At the back, the end-plate is flanged to form the joint with the shell.

The two furnace-tubes are 2 feet 9½ inches in diameter, and about 7 feet in length. They are made of two welded rings of ⅜-inch plate, united by Adamson flange-joints. The joint flue, or kidney-shaped flue, into which the burnt gases are delivered, is segmental in section, in the sense that the top and the bottom are both curved, respectively convex and concave, being concentric, or parts of circles struck from a centre below the boiler. By the concavity of the bottom of the flue, a large internal space is provided within the boiler at the bottom, by which inspection, cleansing, and repair are facilitated. The flue, matching the furnace-tubes, is 2 feet 9½ inches high, and 5 feet 9 inches wide. It is traversed and supported by twenty-eight conical water-tubes or pipes, known as Galloway tubes, 5½ inches in diameter inside at the bottom, and 10½ inches at the top. These tubes are identical in form and interchangeable; they are simple and symmetrical, with square flanges, and are made by means of machinery. In practice, it is maintained, it never becomes necessary to remove or renew one of these tubes, as, if kept clean, they last as long as the other parts of the boiler; yet they are joined to the tube with flanges and rivet-joints, which, of course, are exposed to the action of the fire. The conical pipes are closely placed in the oval flue:—in this 24-feet shell there are 11 transverse rows, in which there are three pipes and two pipes alternately, at a pitch of 19 inches between the rows. The burnt gases as they advance are thereby thoroughly broken up and mixed, and are caused to part with their heat to the water-pipes, as well as to the flue. The circulation of water and steam is thoroughly accomplished. Pockets are formed in the segmental flue,—near the beginning and at the middle,—resembling Petrie's pockets, fig. 274, the design of which is to deflect the lateral streams of gases towards the middle of the flue. The flue is reduced at the back end, by an indentation at each side, similar to the pockets, with the special object of providing elasticity in the back end-plate for the expansion and contraction of the flue when at work; serving also the purpose of checking undue escape of the burnt gases. Elasticity is further secured by the form of the angle-irons joining the ends of the furnace-tubes and the kidney-tube to the shell. These are of uniform thickness at the angle and the webs,—resembling in section a piece of plate bent to a right angle, and curved at the angle.

The ends of the boiler are stayed to the shell by five gusset-plates, with double angle-irons, at each end, the central gusset being riveted to the second ring of plates from each end; and the others to the first rings of plates. They are, as shown in figs. 4 and 5, Plate III., trimmed so as to clear the flues, with an interspace of 8½ inches or 9 inches between the rivets of the gussets and the flues in the end-plates. At the sides of the furnace-tubes, it is true, where they approach nearest to the shell-plates, there are only 4½ inches of interspace between the rivets. This limited allowance is complemented by the wider interspace, 9½ inches, in the back end-plate provided by the pocketing of the kidney-flue, already noticed. Such special provision for expansion has been proved to be beneficial in preventing the grooving of the end-plates. It is the more necessary, since the transverse seams of the kidney-flue are simple lap-joints.

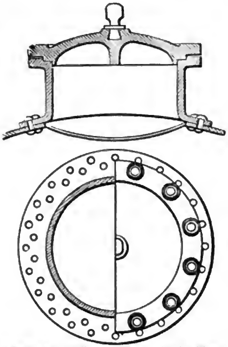
The boiler is tested by a hydraulic pressure of 100 lbs. per square inch. The fire-grates are 5½ feet long, made up of three lengths of cast-iron fire-bars, supported on the dead-plate and on cast-iron bearers, as in figs. 282, and fig. 2, Plate III. The bridges at the ends of the furnace-tubes are fire-lumps, the tops of which are 12 inches below the crowns of the tubes. The furnace fronts are of ½-inch wrought-iron plate, secured to the flue by three eye-bolts and nuts, as in figs. 282, and trimmed with brass beading. The furnace mouth is of cast iron, with the dead-plate, 8½ inches deep, bolted to the furnace front. The doorway is 12½ inches high and 17 inches wide. The space between the furnace mouth and the furnace-tube in which it is placed is filled with fire-brick. The fire-door is of cast iron, perforated with ventilator openings, controlled by a circular brass grid, for the due admission of air as required, or for shutting it off. The door is formed



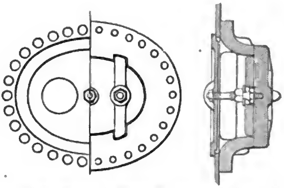
Figs. 282.—Galloway Boiler: Furnace Front, Fire-door, and Bars. Scale 1/16th.

with a hood or baffle-plate at the inner side, by which the air admitted through the grid is deflected downwards towards the grate.

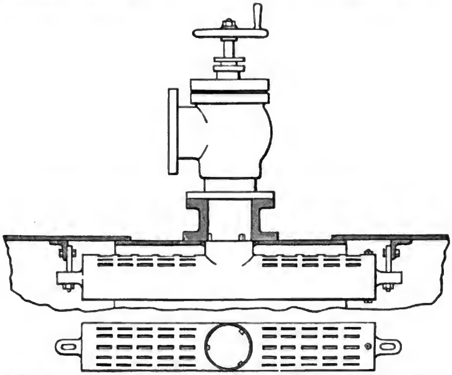
One of Allen's fusible plugs is placed on the top of each furnace. One circular manhole mouthpiece, 17 inches in diameter, figs. 283, is fixed on the top of the boiler, on the second ring of plates from the back end of the boiler. The body is constructed of wrought-iron plate, ¾ inch thick, welded and flanged, and double-riveted to the shell-plate. The cover is of cast iron, 1⅛ inches thick, concave in form, turned to fit the mouthpiece and bolted to it with twelve 1-inch bolts and nuts. There is another manhole, figs. 284, of smaller dimensions, in the front end-plate, near the bottom, serving also the purpose of a mud-hole. It is oval in form, having



Figs. 283.—Galloway Boiler: Circular Manhole and Mouthpiece. Scale 1/16th.



Figs. 284.—Galloway Boiler: Front Manhole. Scale 1/16th.



Figs. 285.—Galloway Boiler: Shut-off Valve and Collecting Steam-pipe. Scale 1/20th.

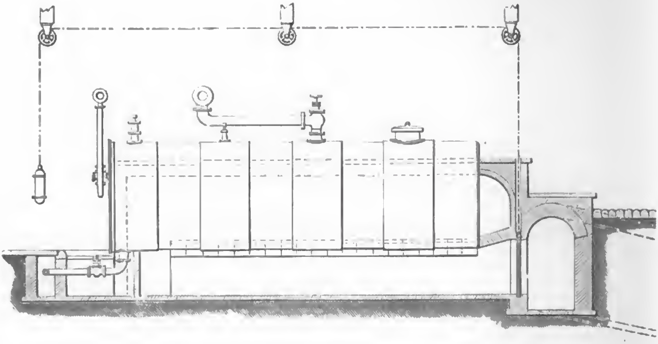


FIG. 1.—SECTIONAL ELEVATION OF BOILER, FLUES, AND CHIMNEY.
Scale 1-96th, or ⅛ inch = 1 foot.

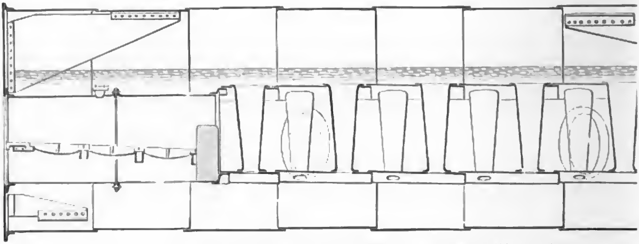


FIG. 2.—LONGITUDINAL SECTION. Scale 1-48th, or ¼ inch = 1 foot.

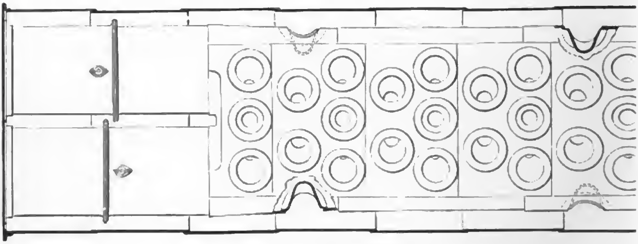
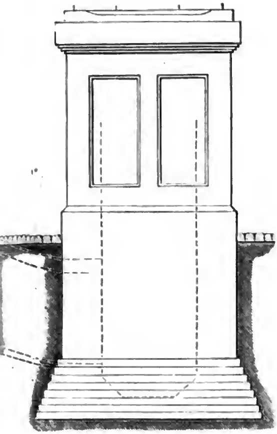


FIG. 3.—SECTIONAL PLAN. Scale 1-48th.

THE GALLOWAY STEAM BOILER,

CONSTRUCTED BY MESSRS. GALLOWAY AND SONS, MANCHESTER.



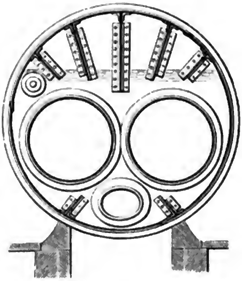
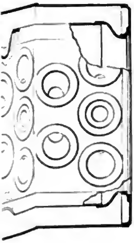


FIG. 4.—TRANSVERSE SECTION THROUGH FURNACE TUBES.
Scale 1-48th.

Diameter of each Boiler, 7 ft.
Length of „ „ 24 ft
Diameter of Furnace Tubes, 2 ft. 9½ in.
Shell Plates, iron, 7/16 in. thick.
Plates of Furnace Tubes, ⅜ in. thick welded.
28 Conical Water Tubes.
Length of Fire-grates, 5½ feet.
Weight with Fittings, 13¼ tons.
Area of Fire-grates, 30.8 sq. ft.
Nominal Horse-power, 50 H.P



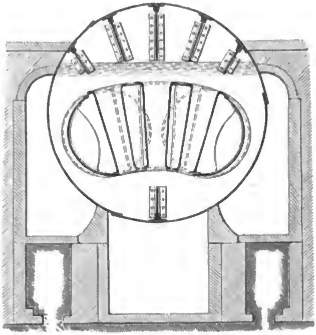
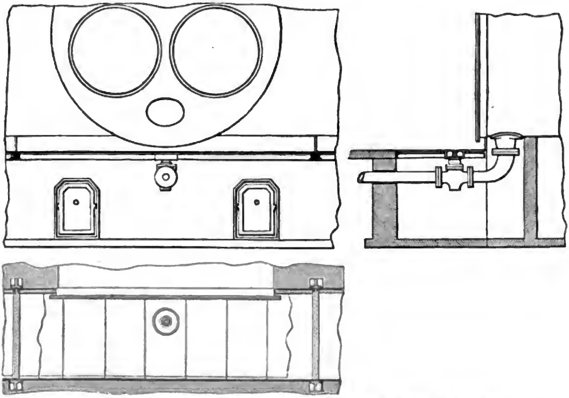


FIG. 5.—TRANSVERSE SECTION THROUGH FLUE TUBE.
Scale 1-48th.

an opening 15 inches wide and 11 inches deep. It is a cast-iron block, placed inside and riveted to the end-plate, with a cast-iron cover applied on the inside, and fastened by two cross-bars and bolts and nuts. A loose cover is put on outside, to finish flush with the end-plate of the boiler.

The steam stop-valve, 6 inches in diameter, and the "anti-priming" collecting pipe, 4 feet long, are shown in figs. 285. There are fifty perforations, 2½ inches by ¼ inch, in the pipe. The steam stop-valve is fixed upon the stand-pipe, into which the steam from the anti-priming pipe is delivered.

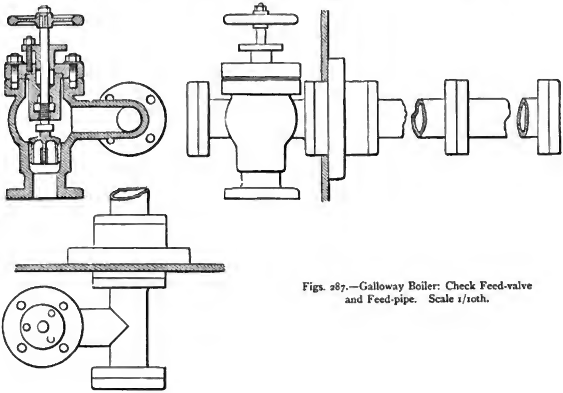
The blow-off cock and elbow-pipe attached to the bottom of the boiler at the front, are shown in figs. 286. The cock is 3 inches in diameter. It is of gun-metal, with a solid or close bottom and a packed gland. By the elbow-pipe the cock is joined to the stand-pipe.



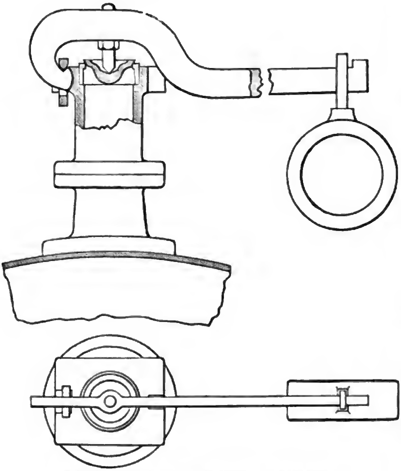
Figs. 286.—Galloway Boiler: Blow-off Cock and Flue Frames and Doors. Scale 1/48th.

The boilers in a range are supplied from a transverse feed-pipe overhead, level with and parallel to the main steam-pipe, from which a vertical branch is taken to each boiler, as shown in fig. 1, Plate III. The check feed-valve, next the boiler, 2 inches in diameter, figs. 287, is loose from the spindle, so as to act as a check or non-return valve, and to be set down by means of a hand-wheel and screw. The feed-pipe is carried within the boiler horizontally for a length of 5 feet; to which is connected an additional pipe, 6 feet in length, perforated with small holes at the lower side, for distributing the feed-water.

One 4-inch lever safety-valve, figs. 288, is fixed on a stand-pipe at the fore end of the boiler. The valve, shown in section, is hollowed so as to give a bearing to the stud-pin at a level below that of the bearing edge of the valve seat. Thus, the valve answers nimbly to excess of pressure and rises. The fulcrum of the lever is a knife-edge, 3½ inches from the centre of the bearing stud, and 37 inches from the end of the extreme range of the lever—making a maximum leverage of about 10 to 1.



Figs. 287.—Galloway Boiler: Check Feed-valve and Feed-pipe. Scale 1/10th.

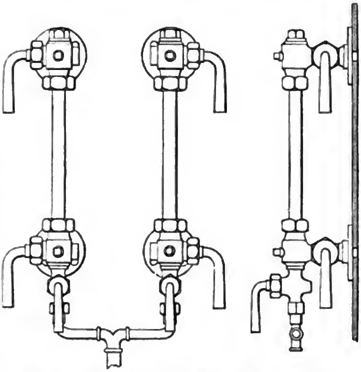


Figs. 288.—Galloway Boiler: Safety-valve. Scale 1/10th.

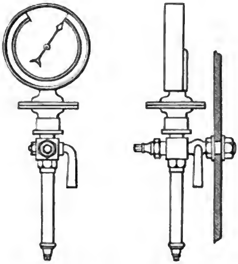
A high-steam and low-water safety-valve also is supplied. Glass water-gauges, consisting of duplicate brass fittings and glass tubes, fixed with india-rubber washers, are shown in figs. 289.

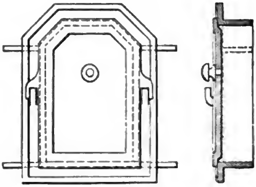
A 7-inch steam-pressure gauge, with brass fittings, is shown in figs. 290.

The type of seating for the boiler is shown in figs. 4 and 5, Plate III.; similar to what has already been described. It is observable that a pair of gussets are placed in the lower part of the shell, at the front, directly over the lumps on which the boiler is supported. A single gusset is fixed centrally at the back. The side flues have a minimum width of 9 inches at the upper part; and the bottom flue is 3 feet deep and 3 feet 3 inches wide. The flue-frame, figs. 291, shown also in figs. 286, of cast iron, is built into the brickwork of the boiler at the ash-pit. The door is readily unlatched and removed when required. The opening is 18 inches high and 12 inches wide.


Figs. 289.—Galloway Boiler: Glass Water-gauges. Scale 1/10th.

The draught, after passing through the flue-tube, is split and returned by the side flues to the front, where it descends and passes by the bottom flue to the far end, and thence to the chimney. It is stated that this system of conducting the burnt gases along the sides first, and afterwards along the bottom, is found to answer much better with the Galloway boilers than the reverse order of passage, under the bottom first. A cast-iron


Figs. 290.—Galloway Boiler: Steam-pressure Gauge. Scale 1/10th.


Figs. 291.—Galloway Boiler: Flue-frame and Door. Scale 1/10th.

damper in a cast-iron frame is placed at the outflow from the bottom flue; balanced by a counterweight connected with it by a chain supported on pulleys overhead, as shown in fig. 1, Plate III.

The radial pipe, or spring steam-pipe, shown in fig. 1, Plate III., is 6 inches in diameter, and 6 feet 10 inches in horizontal length. It is turned upwards at the end to join the main steam-pipe, which is 9 inches in diameter, to receive the steam from four boilers. The shut-off valve, to which the radial pipe is connected, and the perforated collecting steam-pipe within the boiler, are shown in section, figs. 285.

The weight of the boiler, 7 feet in diameter, and 24 feet in length, of 7/16-inch plates, with manhole and steam-pipes, is 11¼ tons. The fittings weigh about 2½ tons additional, making the gross weight of 13¾ tons, or 11⅔ cwts. per lineal foot.

The area of each fire-grate is $(5.5 \times 2.8 =)$ 15.4 square feet; making for two fireplaces 30.8 square feet. The heating surface is: in the furnaces, 70 square feet; in the flue and water tubes, 185 square feet; and on the shell, 290 square feet; together, 545 square feet, which is 17.7 times the grate-area.

The price of the boiler is £355, or about £26 per ton.

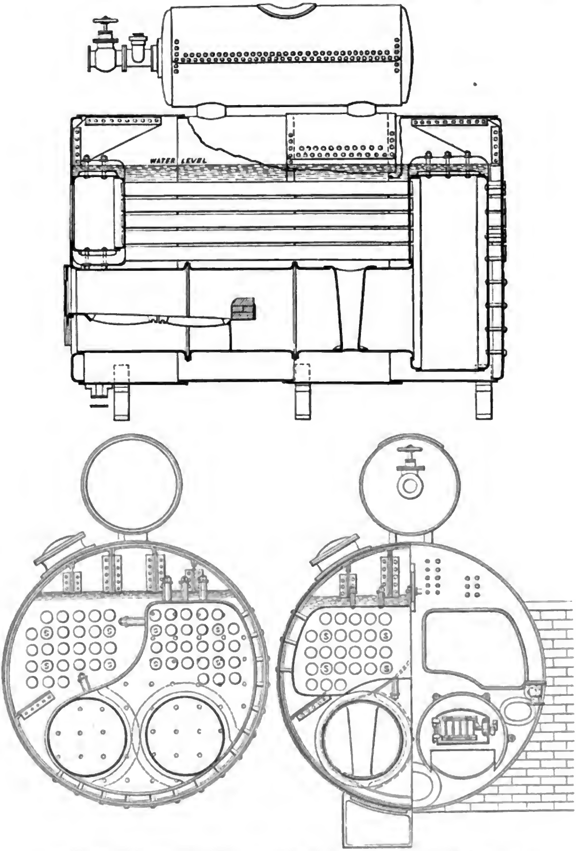
The boiler is of 50 nominal horse-power. The customary allowances made by Messrs. Galloway are from ⅝ square foot to ¾ square foot of grate-area per nominal horse-power, for larger and smaller boilers respectively; and 15 square feet of heating surface per nominal horse-power. In the example now under consideration, ⅝ square foot of grate is allowed, and 11 square feet of heating surface per nominal horse-power. The economy of fuel effected by the introduction of water-pipes into plain flue-tubes, is estimated at from 10 per cent to 25 per cent.

CHAPTER XII.—STATIONARY MULTITUBULAR BOILERS.

The first object in the designing of stationary multitubular boilers is to combine maximum evaporative power with a minimum of floor space or ground space. The small tubes are placed in the line of the return-draught, and are known as return-tubes. The flues are thus in a manner doubled on themselves; and considerable length of internal flue is obtained on a comparatively short length of boiler.

I. MULTITUBULAR STEAM BOILER, BY S. HODGE & SONS.

The return tubular boilers of Messrs. S. Hodge & Sons, figs. 292, are self-contained, within a cylindrical shell. There are two furnace-tubes, containing each a fire-grate; the burning gases from which pass into and meet in a combustion-chamber at the back, where they rise and pass into a group of return-tubes at the left-hand side of the boiler, through which they return to the front, and are delivered into a smoke-box over the furnace front. From this they repass through a group of flue-tubes at the right hand to the back end of the boiler, whence they escape into the flue leading to the chimney. In this way the burnt gases traverse the boiler



Figs. 292.—Return Multitubular Steam Boiler, by Messrs. S. Hodge & Sons, London. Scale 1/48th.

three times—once through the furnace-tubes, and twice through small tubes, all within the shell, and free from brickwork setting. A cylindrical steam-chest or reservoir is attached to the top of the boiler. It is essential that there should be a steam-chest to supplement the steam-space in the shell. The whole of the boiler is of iron, excepting the internal tube-plates and the furnace-tubes, which are of Siemens steel.

The nominal horse-power of the boiler is reckoned on the following bases:—One cubic foot of water evaporated per hour, is one measure of a nominal horse-power; ⅜ square foot of grate-area per horse-power, or 17 square feet of heating surface per horse-power.

The boiler which forms the subject of the illustration, is of 60 nominal horse-power. The shell is 8 feet 7¾ inches in diameter, and 14 feet ½ inch long,—in four rings of plates 9/16 inch thick, single-riveted at the circular seams, and double-riveted longitudinally. The end-plates are ⅝ inch thick and are flanged to join the casing, and also the front ends of the furnace-tubes. They are stayed by four gussets of ½-inch plate at the upper part at each end, riveted in with a pair of 3-inch angle-irons; and by two gussets at the front end, in the spaces between the furnace-tubes and the flue-tubes. The shell-plates are single-riveted with ⅞-inch rivets in 15/16-inch holes, at 2½ inches of pitch, with 2⅝ inches of lap; and they are double-riveted with a longitudinal pitch of 3 inches, and 2 inches of spacing between the two rows of rivets, which are placed zigzag, with 4⅝ inches of lap. The rivet-holes are punched; and the riveting is by machine.

The furnace-tubes are 2 feet 8 inches in diameter internally, and 11 feet in length, measured to the front, in three rings of ⅜-inch plates, united by flange-joints. The back ring is traversed by a vertical conical water-pipe, 5½ inches in diameter at the lower end, and 11 inches at the upper end, riveted to the tube. The front end-plate of the shell is flanged inwards to take the tube; whilst, at the back, the tube itself is flanged to take the tube-plate of the combustion-chamber, lapping behind it, and acting as a stay for it.

The combustion-chamber is made of 7/16-inch plates, except the tube-plate, which is ½ inch thick. It is 2½ feet long internally. The smoke-box, at the front, is of 7/16-inch plate, with a ⅝-inch tube-plate. It is 20½ inches long.

The first group of flue-tubes, 26 in number, are 5⅛ inches in diameter externally, No. 8 wire-gauge, or barely 3/16 inch, thick; and 9 feet 4 inches in length, measured to the outside of the tube-plates. They are pitched at 6¼ inches between centres. The second group of tubes, also 26 in number, are ¼ inch less in diameter than the first, or 4⅞ inches, as, the gases being cooler in the second group, it is thought that they require less space. The tubes are 12 feet 4 inches long, extending to the back end-plate of the shell; pitched at 6 inches between centres. Clear interspaces of 1⅛ inches are left between the tubes. In each group, four of the tubes are fixed as stay-tubes, being screwed into the tube-plates, for which purpose one end of each tube is enlarged to a thickness of 5/16 inch, so that the smaller end may be passed through the larger opening. The other tubes are fixed into the tube-plates by expanders, by which they take the indented form shown in fig. 293.

The combustion-chamber and the smoke-box are stayed to the shell of the boiler, and form with it water-spaces 3 inches wide, about the combustion-chamber, and about 5 inches wide at the back of it. At the smoke-box, the side water-spaces are 4 inches wide. The stay-bolts are pitched at 9 inches apart at the combustion-chamber, and 10 inches at the smoke-box. The other portions of these chambers are partially formed circularly for the resistance of pressure, and the flat portions are stayed with cross-bars externally.

The steam-chest or drum, on the top of the boiler, is 3 feet 3 inches in diameter, and 8 feet long, externally; of ⅜-inch plates, and 7/16-inch plates for the ends. The ends are dished, with a rise of 1½ inches; and are flanged to and riveted to the case, by single-riveting, for which ¾-inch rivets in 13/16-inch holes, at 2 inches of pitch, with 2¼ inches of lap, are employed. The case is formed of one plate bent and double-riveted, with a 2½-inch pitch longitudinally, 2 inches of spacing, and 4¼ inches of lap. The steam-chest is connected to the shell of the boiler by two neckings of cast iron, 6 inches in diameter internally.

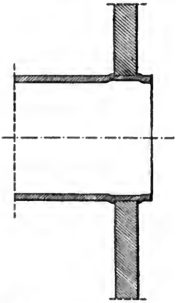


Fig. 293.—Hodge's Boiler: Fastening of Flue-tubes by Expander. Depression caused by Rollers. Scale 1/5th.

The water-level is 4 inches above the crown of the smoke-box; there are about 370 cubic feet of water-room, and 200 cubic feet of steam-room.

There are five manholes in the boiler:—one, 12 inches by 16 inches, at the upper part of the shell, one, of the same sizes, in the top of the steam-chamber, and one, of the same sizes, at the lower part of the front end-plate. There are two smaller manholes in the front plate, 8½ inches by 11 inches, to the right and the left, passable by a small boy.

The furnaces are made with the usual cast-iron front, door, and dead-plate; shown in figs. 292; with cast-iron fire-bars, ⅞ inch thick, and ⅜ inch apart; in two lengths, making each grate 4 feet 8 inches long. The bars are notched upon a central bearer, the other ends of the bars being free to slide on inclined bearing surfaces, as they expand and contract. They have a fall towards the bridge of 2¼ inches in the whole length. Each fire-door is formed with a slide, which is opened for the admission of air above the fire, by a self-acting lever, which is in one piece with the latch, and is moved laterally when the latch is raised for opening the door. The slide is closed

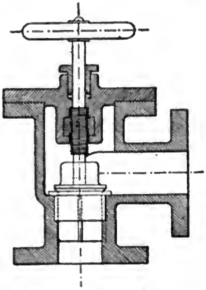
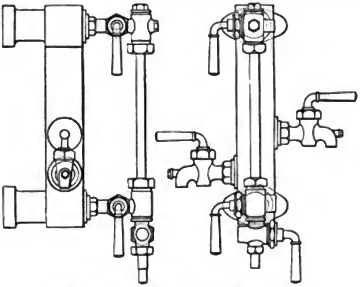


Fig. 294.—Hodge's Boiler: Feed Check-valve. Scale 1/6th.

by hand when required. The bridge is of brickwork on a cast-iron seat. The top of the bridge is 11 inches below the crown of the furnace-tube.

The other fittings comprise dead-plates, steam stop-valve, two safety-valves, feed check-valve, blow-off cock, water-gauge and gauge-cocks, pressure-gauge, safety-plugs, and cradles.

Detail illustrations of the mounting are shown by figs. 294 to 299. The feed check-valve, fig. 294, 2 inches in diameter, rises and falls in a brass seat fixed in the delivery-pipe, bolted to the front of the boiler at the right-hand side. The lift is ⅜ inch, adjusted by means of a screw and hand wheel. The water is delivered direct into the boiler, without any means of dispersion. The water-gauge, figs. 295, is provided with a glass tube to show the level of the water in the boiler, with the needful taps and cleaning plugs, and with two ¾-inch gauge-cocks with cleaning plugs. A 4-inch steam stop-valve, fig. 296, with a 4-inch steam-pipe, is fixed to the steam-chest at one end. It is of cast iron, ¾ inch in thickness. Two lever safety-valves, 3 inches in diameter, fig. 297, are planted on the steam-pipe, between the steam-chest and the steam stop-valve. The blow-off cock, with a 2-inch water-way, fig. 298, is fixed to the bottom of the boiler, at the front. Two of Bailey's safety fusible plugs, fig. 299, are screwed into the crown plates of the combustion-chamber and smoke-box. A plate of fusible metal is placed on the top of the brass seat, and secured to it by a union


Figs. 295.—Hodge's Boiler: Glass Water-gauge.

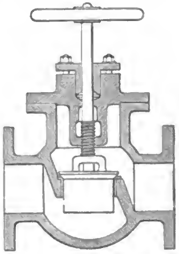


Fig. 296.—Hodge's Boiler: Steam Stop-valve. Scale 1/11th.

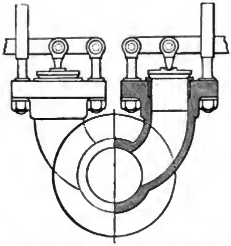


Fig. 297.—Hodge's Boiler: Safety-valves. Scale 1/11th.

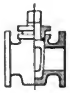


Fig. 298.—Hodge's Boiler: Blow-off Cock. Scale 1/11.

screw. The opening in the seat covered by the metal is 1 inch in diameter, and the plug is 2½ inches high above the plate.

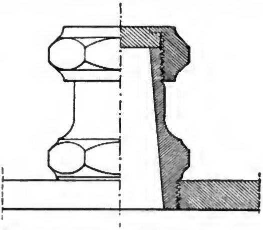


Fig. 299.—Hodge's Boiler: Fusible Plug. Scale ½.

The boiler is supported on three cast-iron brackets, which rest on the foundation, which is of brick or of concrete. The brackets are 4½ feet wide, and 6 inches long. The boiler is laid about ½ inch off the level, the front end being the lower, in order that the whole of the water may be blown off, when required. The shell and receiver are coated with non-conducting composition; or, instead of this, the boiler is inclosed in brickwork, as in an oven, as partly shown in fig. 292.

The area of combined grate-surface is, say, 25 square feet. The heating-surface is 1050 square feet, and is 42 times the combined area of the fire-grates. The area over each bridge is 1.36 square feet, or $\frac{1}{9.2}$ part of the grate-area. The flueway through the flue-tubes is, for the first group, 3.20 square feet, or ⅛th of the grate-area; and through the second group, 2.87 square feet, or $\frac{1}{8.7}$ of the grate-area.

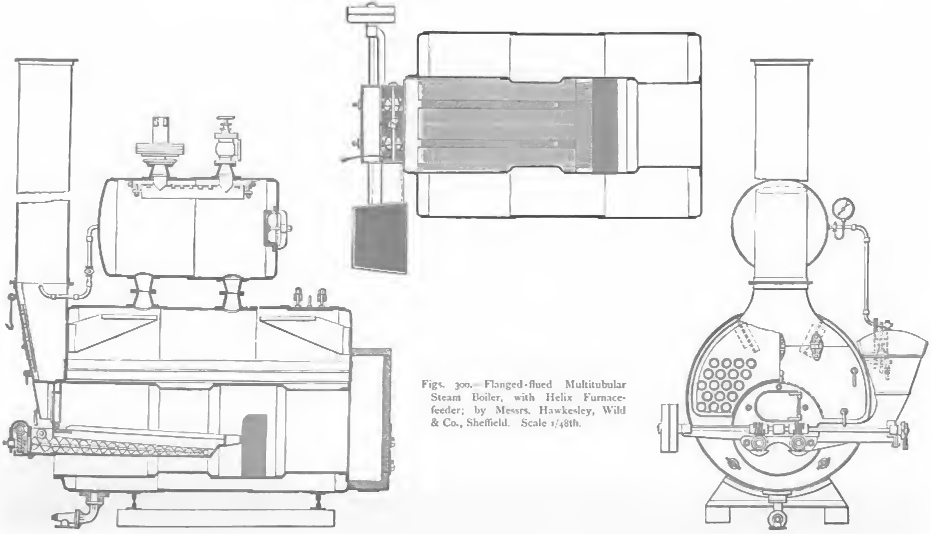
The fire-brick flue constructed at the back of the boiler is 3 feet wide and 2 feet 8 inches deep at the boiler, and thence is reduced in width to 1 foot 6 inches, having a sectional area of 4 square feet, or $\frac{1}{6.25}$ part of the grate-area. By soot doorways in the brick flue, access is had to the ends of the flue-tubes for inspection and repair. The chimney should be 70 feet in height above the level of the fire-grates, and 2½ feet in diameter at the top, having a sectional area of about 5 square feet, or 1/5th of the area of fire-grate. The boiler weighs, without fittings, 13 tons, and with fittings, 14 tons. The working pressure of steam is 60 lbs. per square inch.

After an experience of upwards of 12 years, of the class of boiler here exemplified, internal grooving, it is stated, has not yet appeared.

A 60-horse-power boiler, with a good draught, consumes Welsh coal economically, it is stated, at rates up to 20 pounds per square foot of grate per hour, or totally (25 × 20 =) 500 pounds per hour; evaporating from 9 pounds to 12½ pounds of water from the temperature 120° F. per pound of coal, or from 70 to 100 cubic feet per hour.

II. FLANGED-FLUED MULTITUBULAR BOILER, BY HAWKESLEY, WILD & CO.

The multitubular boiler of Messrs. Hawkesley, Wild & Co., figs. 300, is, like Messrs. Hodge's boiler, self-contained; but it differs in many respects. Within a cylindrical shell, a single furnace-tube is placed, and is flanked by



Figs. 300.—Flanged-flued Multitubular Steam Boiler, with Helix Furnace-feeder; by Messrs. Hawkesley, Wild & Co., Sheffield. Scale 1/48th.

return flue-tubes on each side. A steam-receiver is fixed on the top. The draught, after passing through the furnace-tube, is received in a chamber of fire-brick at the back, from which it returns by a split action through the tubes at both sides together, and reunites in a smoke-box at the front, above the fire-door, whence it passes upwards, by an outflow 2 feet in diameter, into the chimney. The unit of nominal horse-power is the evaporation of 1 cubic foot of water per hour.

The illustrations, figs. 300, represent a boiler of 30 nominal horse-power, fitted with the Helix Furnace-feeder. The plates are entirely of steel. The shell is of plates $^{13}/_{32}$ inch thick (⅜ inch + $^{1}/_{32}$ inch); except the end-plates, which are ½ inch thick; 5 feet 9 inches in diameter outside, and 9 feet in length; stayed with two gussets at each end, at the upper part. The transverse seams are single-riveted, and the longitudinal seams are double-riveted. There is a manhole on the top of the shell, 11 inches by 15 inches. The border of the hole is strengthened by a ring of metal riveted to it. There are two mud-holes and doors, 5 inches by 3 inches, in the front end-plate. The steam-receiver is 3 feet in diameter, and 5½ feet long, of $^{5}/_{16}$-inch plates for the barrel, and one end is dished, with a rise of 3 inches. The other end is flat, and is formed to receive a manhole frame and door. The receiver is joined to the boiler-shell by 4-inch wrought-iron neckings.

The furnace-tube is constructed of steel plates $^{11}/_{32}$ inch (⅜ inch less $^{1}/_{32}$ inch) thick, on the flange-flue system already described, page 689, consisting of four rings, in 2-feet lengths, except the front ring, which is longer, alternately of larger and smaller diameters:—in the present instance, 3 feet and 2 feet 8 inches, showing a flanging of 2 inches at each side. The return-tubes, 18 at each side, or 36 at both sides, traverse the shell from front to back. They are 3½ inches in diameter outside; placed at a pitch of 4¾ inches between centres, and showing a clearance space of 1¼ inches between the tubes.

The working pressure is 80 lbs. per square inch. To evaporate 30 cubic feet of water per hour, corresponding to 30 nominal horse-power, the use of the steam-jet in the chimney is required.

The fire-grate is 5½ feet in length, with a fire-brick bridge; it has an area of 15 square feet. The total heating surface of the boiler amounts to 375 square feet, or 25 times the grate-area. The upper surface of the bridge is 11 inches below the crown of the furnace, forming a sectional area of way over the bridge, of 2.10 square feet, or $^{1}/_{7}$th of the area of fire-grate. The flueway through the flue-tubes is, for both sides together, 2.08 square feet, or $^{1}/_{7}$th of the grate-area. The upward flue or chimney exit, has a sectional area of 2.10 square feet, or $^{1}/_{7}$th of the area of fire-grate; being the same as the area of way over the bridge and through the tubes.

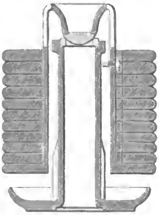


Fig. 301.—Hawkesley, Wild & Co.'s Boiler: 2¼-inch dead-load Safety-valve. Scale 1/10th.

The boiler is supported on two plate-brackets fastened to sleepers.

A 3-inch steam stop-valve is fixed on the top of the steam-reservoir; also a 2¼-inch direct-load safety-valve, fig. 301. A perforated collecting pipe, 4 inches in diameter and 3¼ feet long, of cast iron, is slung from the top of the steam-reservoir, provided with two outlets, one to the steam stop-valve, and the other to the safety-valve. A 2-inch blow-off tap is fixed to the bottom of the boiler at the front.

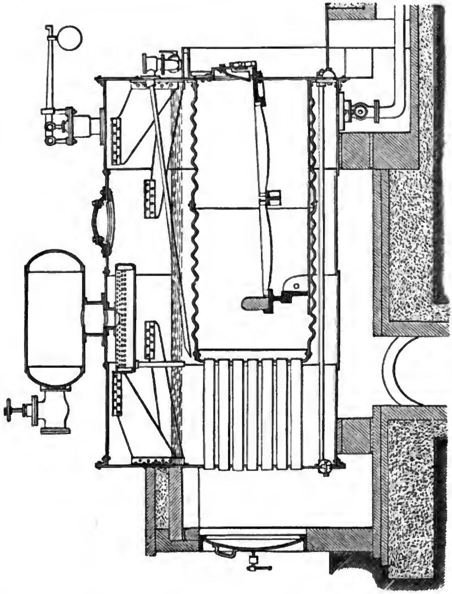
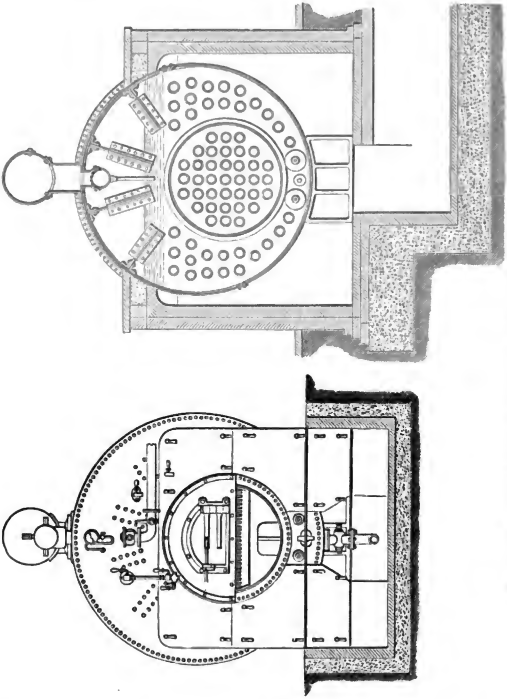


Fig. 302.—Multitubular Return-tube Steam Boiler, by Mr. John Bellamy, London. Scale 1/48th

The furnace is fitted with Holroyd-Smith's Helix Furnace-feeder, or mechanical stoker. This apparatus, in its original form, has already been described, page 338. In the present instance, the hopper is placed at the right-hand side of the smoke-box, the slack descending into a trough laid transversely across the front of the boiler, just below the level of the dead-plate. In this trough it is moved across the furnace mouth by a feeding screw at the bottom of the hopper, 5¼ inches in diameter. Thence it is fed into the furnace by two longitudinal screws in two troughs, below the level of the fire-grate, where it is at the same time lifted by the action of the screws and deposited on the grate to the right and the left. The longitudinal screws are double-threaded, 5 inches in diameter at the front, tapering to 2 inches in diameter at the far ends. The pitch of the threads at the commencement is 8 inches, and is gradually reduced to 6 inches at the far end. The screws are moved by worms on a transverse shaft, turned by rachet-wheel and pawl movement, and make from 2 to 3 revolutions per minute.



Figs. 303.—Multitubular Return-tube Steam Boiler, by Mr. John Bellamy, London. Scale 1/48th.

The total weight of the boiler, including all mountings, except the furnace-feeder, is 5 tons 14 cwts. The net weight of the boiler and chimney alone is about 5 tons.

III. MULTITUBULAR BOILER, BY JOHN BELLAMY.

The type of boiler manufactured by Mr. John Bellamy, figs. 302 and 303, though constructed as a self-contained boiler, is set in brickwork, and is surrounded by external supplementary flues. The shell is cylindrical, surmounted by a steam drum or chest, with a single internal furnace-tube, from which a multitubular flue extends to the back end of the boiler. The draught is split at the back and returns through a number of small flue-tubes, at each side of the furnace-tube, to the front into a smoke-box of iron plate, whence it descends and is conducted under the boiler to the chimney-flue at the back. In this way, the boiler is traversed three times by the current of burnt gases before leaving for the chimney. For one nominal horse-power, 18 square feet of heating surface within the boiler is allowed, not reckoning the external heated surface of the shell, which is additional. Half a square foot of fire-grate area is allowed for one nominal horse-power. A maximum evaporation of 5 pounds of water per square foot of heating surface per hour is estimated. For economical work, only 4 pounds per hour is estimated, when an evaporation of 10 pounds per pound of coal is assumed. From these quantities, it follows that (18 × 5 =) 90 pounds maximum, or, for economical evaporation, (18 × 4 =) 72 pounds, or a little over 1 cubic foot, of water is evaporated per horse-power per hour. Per square foot of grate, (72 × 2 =) 144 pounds, or 2.31 cubic feet, of water are economically evaporated per hour; and (144 ÷ 10 =) 14½ pounds of coal are consumed per square foot of grate per hour. It is stated that the temperature of the burnt gases in the front smoke-box seldom exceeds 400° F.

Particulars of dimensions and heating surfaces of boilers of various powers are given in the annexed tablet:—

Nominal horse-power	4	6	8	10	12	14	16	20	25	30	35	40
	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.
Length	5 3	6 3	6 7	7 2	7 8	7 9	8 2	8 9	9 6	9 8	12 1	12 9
Diameter	3 3	3 9	4 0	4 4	4 8	5 1	5 4	5 10	6 0	7 0	7 3	7 6
	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.
Internal heating surface	72	108	145	180	216	252	288	361	449	540	637	737

The boiler selected for illustration is one of 40 nominal horse-power. The shell of the boiler is 7½ feet in diameter, and 12 feet 9 inches long, of 9/16-inch iron plates, except the ends, which are each in one plate 11/16 inch thick. The ends are riveted to the shell by 3-inch angle-iron welded rings externally to the shell; and they are stayed to the body by four gussets of ⅝-inch plate at each end, riveted with pairs of 3-inch angle-irons. There are, in addition, below the furnace-tube, two through stay-bolts, 1¾ inches in diameter, fastened with double nuts and washers at each end. The longitudinal joints in the shell are double-riveted; the transverse seams are single-riveted. The boiler is constructed for a working pressure of 80 lbs. per square inch, and is tested by hydraulic power to a pressure of 120 lbs. per square inch.

The steam is taken into a perforated pipe previously to its rising into the drum. The pipe is 8 inches in diameter, and 3½ feet in length, numerously perforated with ½-inch holes. It is fitted with a trap-pipe which descends below the water-level, through which any water that may be drawn into the pipe by the steam falls back into the boiler. The steam stop-valve is fixed to one end of the drum.

A manhole piece is formed out of a ⅞-inch plate, to a diameter of 12 inches, with a dished wrought-iron cover, 1 inch thick, secured with twelve 1-inch holding-down studs and nuts.

The steam-drum is of ⅜-inch plate, 21 inches in diameter, and 4½ feet long, with dished ends, having a rise of 2½ inches. It is joined to the boiler-shell with a wrought-iron welded neck 12 inches in diameter, riveted to the boiler. But the opening through the shell is only 6½ inches in diameter.

The furnace-tube is of Fox's corrugated plate, 3 feet 10 inches in medium diameter, 9 feet 2 inches long, of ⅜-inch mild steel plate, except the tube-plate, which is 9/16 inch thick. The tube-plate is flanged to join the tube, and the tube is joined to the front plate of the shell with a 3-inch angle-iron ring. The flue-tubes direct from the furnace-tube to the back end of the shell, 37 in number, are 4½ inches in diameter, No. 9 wire-gauge, or about 5/32 inch in thickness, and about 3½ feet long between plates. There are 32 return-tubes at the side of the furnace-tube, 5 inches in diameter, No. 8 wire-gauge, or nearly 3/16 inch in thickness, and about 12 feet 7½ inches long between the end-plates of the shell.

All the edges of the plates are planed, and the edges of the end-plates turned. The openings for the furnace-tubes also are turned out. The rivet-holes, usually, are punched; but they are drilled when required. In cases where the whole of the boiler is made of mild steel, the holes are drilled in place. The rivets are 13/16 inch in diameter, at 2 inches of pitch, with 2¾ inches of lap, for single-riveted seams; and at 3-inch straight pitch, and 2-inch zigzag pitch, with 4¾ inches of lap for double-riveting, in the longitudinal seams.

The flue-tubes are fixed by expanding in the usual manner; those in the fire-box are beaded. No special stay-tubes are employed.

The furnace is fitted with a wrought-iron plate front, cast-iron furnace-frame, cast-iron door carrying a baffle-plate inside, by which the door is protected, and the air that enters through the door-slides is deflected towards the dead-plate. The dead-plate is 10 inches long; and the grate is 6¼ feet in length, giving an area of 22 square feet, which is at the rate of a little over ½ a square foot per nominal horse-power. The fire-bars are ⅝ inch thick at the upper surface, and 4½ inches deep, with ½-inch air-spaces for ordinary coal. The bridge is a fire-lump on a cast-iron seat. It is 18 inches below the crown of the furnace, and gives a sectional area of passage of 4⅓ square feet, for the burnt gases, or $^1/_5$th of the area of grate. The flueway through the direct flue-tubes is 3½ square feet, or $^1/_{6.3}$ part of the grate-area; through the return-tubes it is 3¾ square feet, or $^1/_{5.9}$ part of the grate-area. The flueway outside the boiler has a sectional area of about 22 square feet, the same as the area of the fire-grate.

The heating surface is reckoned as follows:—In the furnace-tube, taking the whole of the surface excepting the portion below the fire-grate, 67 square feet; direct tubes, 147.3 square feet; return-tubes, 523.6 square feet; making together 737.9 square feet—33.6 times the grate-area, or, say, 18 square feet per nominal horse-power. There is, in addition, an area of 144 square feet, at the sides and bottom of the shell, in the bottom flue, which makes a total of 882 square feet of heating surface;—40 times the grate-area, or 22 square feet per nominal horse-power.

The boiler is supported on two cast-iron stools set in the brickwork.

The mountings include a double 4-inch lever safety-valve, with levers, weights, and counterweights; a 5-inch steam stop-valve, with spindle; a 2-inch brass check feed-valve, fixed to the front of the boiler, from which the water is carried in a tube to the back of the furnace-tube, where it is discharged; a 3-inch brass blow-off cock; a set of ¾-inch water-gauges; a steam-pressure gauge, 7-inch dial; two ¾-inch test-cocks; fusible plug in flue; and internal steam-collecting pipe.

The net weight of the boiler, with the steam-chest, is about 9¼ tons. With furnace fittings, front smoke-boxes, and mountings complete, the weight is 11¼ tons. The price is £512:—at the rate of £45, 10*s.* per ton, or £12, 16*s.* per horse-power.

CHAPTER XIII.—VERTICAL STEAM BOILERS.

Vertical boilers are used chiefly for the sake of compactness and of simplicity of construction and establishment. It appears that they were first employed for utilizing the waste heat from puddling, reheating and other furnaces. They are known as upright furnace boilers; and a large number are at work on this duty. The Rastrick type of vertical boiler, is cylindrical, with hemispherical ends, and a vertical central flue from the lower end, extending upwards for about half the height of the boiler. The flue branches to and through the sides of the boilers by two, three, or four horizontal necks. The boilers are surrounded by brickwork flues, below the water-level. The combustible gases are introduced at the lower part of the boiler, near the bottom, where they burn and impinge on the sides or walls; thence the burning gases pass upwards around the boiler, to the upper part of the external flue; whence they pass into the central flue, and descend to the bottom, where they escape by an underground flue to the chimney. The gases are not delivered uniformly around the boiler, but unequally,—from, two, three, or four furnace-necks. The inequality of heat causes inequality of expansion, and, besides, the plates opposite the furnace-necks are exposed to overheat, with consequent fracture and blister, unless they are protected by bafflers of fire-bricks. Overheating, moreover, is the more likely to take place as the initial temperature of the gases is elevated, and the temperature of combustion is the higher in consequence. Such boilers are from 25 feet to 30 feet in height, and from 8 feet to 10 feet in diameter. These proportions are not favourable for high pressures, as, whilst the diameters are considerable, the thickness of plates is diminished, and double-riveted seams are forbidden by the conditions of external firing. Mr. Fletcher mentions the case of a Rastrick boiler, heated by the combustible gases from four iron furnaces, which exploded. It was 27 feet high, 10 feet in diameter, of plates from $5/_{16}$ inch to $7/_{16}$ inch in thickness. The brickwork of the flue was 15 feet high. The external shell had a rent in the lower part, and an entire ring of it was stripped,—the result of intense flame locally applied, with sedimentary water. The flame struck on a confined water-space, where, probably, the circulation was imperfect. The boiler had had a dose of bran a few weeks previously; it stopped the leaks, but it thickened the water.

In another form of vertical boiler, for utilizing waste gases, the shell is cylindrical, and is usually flat-ended; and it is traversed by a plain central flue. The height of such boilers is usually about 30 feet. The combustible gases are delivered into the flue at the lower end in which they are burned, and through which the burnt gases pass upwards, escaping at the top. It is naturally less efficient than the Rastrick boiler, as the external surface is not utilized for heating. But the plates may, for the same reason, be made of any required thickness, and double-riveted; and, not being externally fired, they are not subjected to unequal expansion. The lowermost ring of plates of the flue should be protected from the direct action of the flame by a lining of fire-bricks; so also should the upper part of the flue, in the steam-space; the flue being enlarged so that the lining may be flush with the lower part of the flue.

Horizontal boilers, also, it may be noted, of the Cornish or the Lancashire type, are used for utilizing waste heat; and, where room is to be had, they are preferred for this purpose to upright boilers. The gases are ignited at the front ends of the flues, which are protected by fire-brick lining; and the current of burnt gases, after having passed through the flues, should be conducted round the outsides of the boiler;—for the sake of economy, and also for prevention of inequality of expansion, and greater durability of the boiler.

Ordinary Vertical Boilers.—The common type of cylindrical vertical boiler for steam-cranes, winches, and miscellaneous purposes, where a comparatively small supply of steam is required, is fairly represented in section by fig. 304. The shell of the boiler is 45 inches in diameter inside the middle ring of plates, and 8 feet in total height above the lower edge of the shell; or with the ash-pit, which is 10 inches high, 8 feet 10 inches above the base. It is made up of three rings of plates, of which the lowermost is extended downwards to form the ash-pit, with an opening in front. The circular seams are single-riveted; and the vertical seams are double-riveted. The fire-box is 3 feet in diameter inside, at the level of the grate, and 2 feet 10 inches at the top, giving 2 inches of taper. It is formed of a single plate, which, after the water-tube holes are cut out, is rolled to form; and the vertical seam is welded up. The water-space is about 3 inches wide at the bottom, and opens to about 4½ inches wide at the top. There are, in the upper part of the fire-box, two transverse water-tubes, 9 inches in diameter, inclined a little, crossing each other; they are welded to the fire-box plate, thus obviating the double thickness of plate required for riveting. Sufficient freedom of circulation of water and steam are thus provided for between the shell and the fire-box, without, by excessive taper, unduly contracting the upper part of the fire-box, which is valuable as heating surface. The crown of the fire-box is 4 feet 3 inches high above the fire-grate. It is convex, having a rise of 3 inches; but the crown of the shell is flat. The two crowns are flanged to join the side-plates, and they are connected by the vertical uptake-flue, 9 inches in diameter inside the necking and the lining of fire-clay. This lining is designed as a protection from the heat, in the steam-space; and to receive it, the flue is enlarged to 12 inches in diameter. Such protection is useful, and should be applied to all such boilers. Instead of a fire-clay lining, some makers insert a sleeve of cast iron.

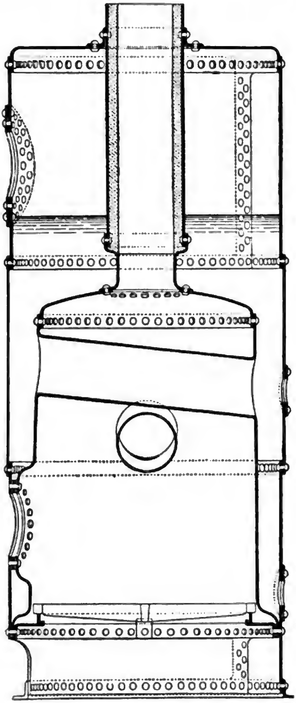


Fig. 304.—Ordinary Vertical Steam Boiler. Scale 1/20.

The crown of the fire-box is much strengthened by its convexity of form, and is in so far fortified against the accident of overheating by shortness of water. The crown is, of course, very rigid; but sufficient elasticity for expansion is provided by the flat crown of the shell. The fire-box is dished or set out at the base, to make the joint with the shell; and also at the firehole, where the dishing is partly relieved by the insertion of a thin ring of metal between the two plates, to complete the joint; providing at the same time, space for the circulation of steam around the doorway. The circular seams are single-riveted, and the vertical seams are double-riveted. There is a manhole in the upper ring of plates, 11 inches in diameter, fortified by a wide stiffening ring, double-riveted to the shell. Mud-holes are placed, one in the bottom of the shell, and one opposite the lower end of each cross-tube.

The fire-bars are of cast iron, supported on a ring-bearer of wrought iron. The area of fire-grate is 7 square feet, and the sectional area of the chimney is .44 square foot, or $^1/_{16}$th of the grate-area. There is 66¾ square feet of heating surface, which is 9.54 times the grate-area.

The crown of the shell is 3¼ feet above that of the fire-box, of which there is 1 foot of water to the normal level, and 2¼ feet of steam-space.

The vertical tubular steam boilers manufactured by Messrs. Daniel Adamson & Co., Dukinfield, are illustrated by fig. 305. They are constructed of from 2 to 12 nominal horse-power, to work at pressures of from 40 lbs. to 150 lbs. per square inch. The unit of power is 13 square feet of heating surface per nominal horse-power. In reckoning the area of heating surface, the tube-surface exposed to the steam is included. The vertical seams of the shell are double-riveted, and those of the fire-box are solid welded. Rivet-holes are drilled, and the edges of the plates are planed.

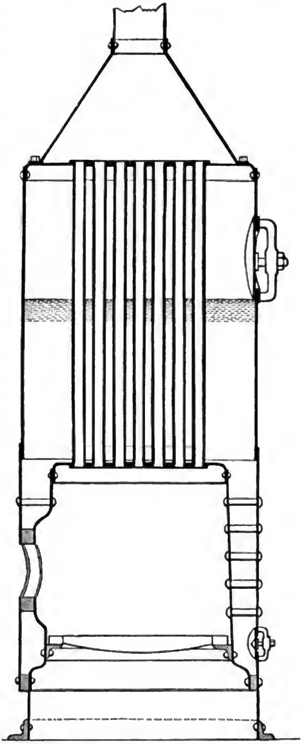


Fig. 305.—Vertical Steam Boiler, by Messrs. D. Adamson & Co., Dukinfield. Scale 1/24th.

In the boiler of 12 horse-power, having a working pressure of 100 lbs. per square inch, the shell is 10 feet high and 4 feet in diameter, of ⅜-inch plates; and the ash-pit is 9 inches high. The chimney is slightly taper, being 14 inches in diameter at the bottom, and 16 inches at the top. It stands 18 feet high above the ground-level. The fire-box is of 13/32-inch plates, 3 feet 5 inches in diameter at the level of the fire-grate, tapering to 3 feet 2 inches at the top. The total height of the fire-box is 4 feet 3 inches. There are 37 flue-tubes, 2½ inches in diameter outside, 5 feet long, fastened into ⅝-inch tube-plates, on which their ends are turned over. The ordinary water-level is at two-thirds of the length of the tubes, or 3 feet 4 inches above the fire-box. The area of fire-grate is 9 square feet, or ¾ square foot per nominal horse-power. There are 150 square feet of heating surface, or 13 feet per horse-power. The ratio of grate-area to heating surface is as 1 to 17.33.

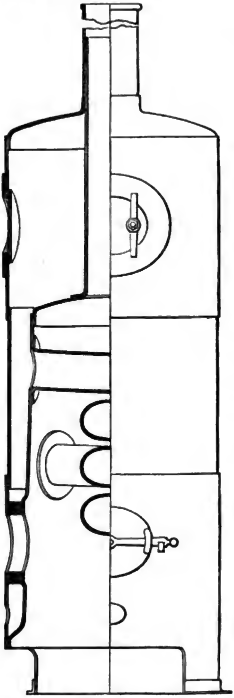


Fig. 306.—Vertical Cross-tube Steam Boiler, by Messrs. Abbott & Co. Scale 1/24th.

The boiler is provided with manhole and cover, hand-holes, uptake, smoke-box, chimney, fire-bars, ash-pit, fire-door, safety-valve, steam-indicator, water-gauges, check feed-valve, stop-valve, and blow-off tap. It weighs, without mountings, 46 cwts.

It is stated that, in normal conditions, 16 pounds of coal are consumed per square foot of fire-grate per hour, or a total of (16×9=) 144 pounds per hour. It is stated also that water is evaporated at the equivalent rate of 8 pounds per pound of coal, from and at 212° F., or (8 lbs. × 16 lbs. × 9 square feet ÷ 62.5 lbs. =) 18.4, or, say, 18 cubic feet per hour; equivalent to 2 cubic feet per square foot of grate per hour, or 1½ cubic feet per horse-power per hour.

Messrs. Alfred Marshall & Co. manufacture vertical boilers of from 1½ nominal horse-power to 12 horse-power; with vertical flue-tubes, and with horizontal cross water-tubes. The 8 horse-power boilers, having vertical tubes, have shells of ⅜-inch plates, 3½ feet in diameter, 9 feet high, and a base or stand 6 inches high. The fire-box is of 7/16-inch plates, 3 feet in diameter, and 3 feet 7 inches high. The tube-plates are ⅝ inch thick. There are 20 flue-tubes, 2¾ inches in diameter outside, 5 feet 5 inches long. The fire-grate has 7 square feet of area, or ⅞ square foot per nominal horse-power; and the heating surface is 114 square feet, or 16.3 times the grate-area, or at the rate of 14¼ square feet per horse-power.

Vertical cross-tube boilers, illustrated by fig. 306, are manufactured by Messrs. Abbott & Co. of from 1 to 16 nominal horse-power, reckoned at the rate of 10 square feet of heating surface per nominal horse-power. The grate-area is a fluctuating quantity, dependent on the diameter of the fire-box, ranging from 1.4 square feet for 1 horse-power to 6.7 square feet for 8 horse-power, or .84 square foot per horse-power; and to 15.3 square feet for 16 horse-power, or nearly 1 square foot per horse-power. For the same nominal powers the heating surfaces are respectively 7 times, 12 times, and 10½ times the grate-area. Thus the grate-area is a resultant of the dimensions of the boiler suited to present given areas of heating surface, rather than a quantity calculated by any fixed ratio.

The working pressure of steam is from 70 lbs. to 80 lbs. per square inch; the test-pressure is 150 lbs. per square inch, hydraulic.

The following selection of data from the practice of Messrs. Abbott & Co., table No. 153, presents in one view the leading dimensions and proportions of their vertical cross-tube boilers.

Table No. 153.—VERTICAL CROSS-TUBE BOILERS FOR LAND PURPOSES.
ABBOTT & CO.

Nominal Horse-power		1	2	3	4	5	6	7	8	9	10	12	14	16
Total height of cylindrical shell	ft. ins.	4 0	5 0	6 0	6 6	7 0	7 6	8 6	8 6	9 0	9 6	11 0	11 0	12 0
Diameter of do., outside..........	" "	1 9	2 6	2 6	2 9	3 0	3 3	3 3	3 6	3 9	4 0	4 0	4 9	5 0
Cross-tubes .....number		1	1	2	2	2	3	3	4	4	4	5	5	5
Do. diameter outside	inches	6	7	7	8	8	8	8	8	9	9	9	9	9
Uptake, outside diameter.......	"	6	7	7	8	8	9	9	10	10	12	12	14	15
Total height of fire-box, excluding crown	ft. ins.	2 5	3 3	3 9	4 0	4 6	4 10	5 1	5 6	6 0	6 9	7 6	7 9	8 6
Thickness of shell ..inch		5/16	5/16	5/16	3/8	3/8	3/8	3/8	3/8	3/8	3/8	3/8	7/16	7/16
Do. fire-box	"	3/8	3/8	3/8	3/8	3/8	3/8	3/8	3/8	3/8	3/8	3/8	7/16	7/16
Do. do. crown............	"	5/16	3/8	3/8	3/8	3/8	3/8	3/8	3/8	3/8	7/16	7/16	7/16	7/16
Heating surface, including uptake	sq. ft.	10	20	30	40	50	60	70	80	90	100	120	140	160
Grate-area..........	"	1.4	3.3	3.3	4	5	6	6	6.7	7.8	9.0	9.0	13.6	15.3
Ratio of grate-area to heating surface.......		7.1	6.1	9.1	10	10	10	11.7	12	11.5	11.1	13.3	10.3	10.4
Net weight of boiler only......	cwts.	6	11	13	17	20	24	27	30	34	37	44	66	80

The gross weight of the 8-horse-power boiler, with fire-bars and mountings, is 33¾ cwts. The prices of these boilers, with chimney and fire-door, and without ash-pit, fire-bars or mounting, vary from £17 for 1 horse-power to £62 for 8 horse-power, and £160 for 16 horse-power. These prices, reduced to a rate per horse-power and a rate per ton net weight of boiler, are as follows:—

	1 Horse-power.	8 Horse-power.	16 Horse-power.
Per horse-power	£17	£8	£10
Per ton net weight	£57	£41	£40

With the extra prices for the ash-pit, fire-bars, and mountings the total price of the boilers complete are, for 1 horse-power, £23, 5*s.*; for 8 horse-power, £75, 15*s.*; and for 16 horse-power, £186; being at the following rates per horse-power:—

	1 Horse-power.	8 Horse-power.	16 Horse-power.
Per horse-power	£23, 5*s.*	£9, 9*s.*	£11, 12*s.*
Per ton gross weight	—	£44, 9*s.*	—

The mountings include the safety-valve, steam stop-valve, feed-valve, water-gauge, 2 gauge-cocks, pressure-gauge, and blow-off cock.

The vertical cross-tube boiler, of 8 horse-power, is constructed of ⅜-inch iron plates, except the crown of the fire-box, which is of steel. The shell is double-riveted in the vertical seams. It is 3½ feet in diameter externally, and 9 feet 3 inches in total height from the ground to the shoulder of the crown-plate. Deducting the height of the ash-pit, 9 inches, the height of the boiler proper is 8½ feet to the shoulder, or, including the camber of the crown-plate, which is dished 4 inches, the height is 8 feet 10 inches. There is a manhole in the upper part, 14 inches by 11 inches, fortified externally by a ring of ¾-inch plate; one mud-hole 3½ inches by 5 inches, opposite one end of each cross-tube; and 3 mud-holes at the base. The fire-box is 2 feet 11½ inches in diameter internally at the level of the fire-grate, and is tapered upwards to a diameter of 2 feet 8 inches at the top. It is usually constructed with two plates, riveted in vertical seams; or it is welded, when required. It is set out at the base, to form the rivet-joint with the shell-plate; but it is joined to the shell with a solid iron ring, at the firehole. The flanging out of the fire-box at the base is preferred for the sake of economy, and of short rivets and good joints. The ring at the firehole is preferred, for the sake of durability, resisting the wear of the fire-irons better than the plate-to-plate joint. With this joint, too, the inner plate is liable to crack by the sudden chill occasioned when the fire-door is left open to keep down the generation of steam.

The water-space is 2½ inches wide at the bottom, and widens to 4 inches at the top. The crown-plate is dished, with a rise of 3 inches. It is united to the crown-plate of the shell, by the uptake-flue, which is of ⅜-inch plate, and is 10 inches in diameter outside, or 9¼ inches inside. The flue is surmounted by a chimney 10 inches in diameter, and 6 feet high above the boiler. The fire-box is fitted with four cross water-pipes, slightly inclined, 8 inches in diameter outside, and 7¼ inches inside, flanged and riveted to the fire-box. The fire-grate is at a level 3 inches above that of the bottom of the water-space, or 6½ inches above the bottom of the fire-box. The fire-bars are ¾ inch thick, with ½-inch air-spaces. The area of fire-grate is 6.70 square feet, and there is 80 square feet of heating surface, including one foot in height of the flue through the upper part of the boiler, equal to 12 times the grate-area. Of this quantity of heating surface, the cross-tubes provide more than one-fourth. The area of flue-way to the chimney is 67 square inches, or .56 square foot; and it is $^{1}/_{12}$th of the grate-area. The water-level is from 6 inches to 8 inches above the crown of the fire-box.

The quantity of water evaporated per pound of coal averages 6 pounds from ordinary temperatures per pound of ordinary coal.

Mr. H. S. Hele Shaw[1] gives the results of experiments on the performance of two small vertical steam boilers, one of which had tubular flues and the other an uptake and a cross tube.

	Tubular.	Cross-tube.
Nominal power	2 H.P.	4 H.P.
Height of shell	4 ft. 6 ins.	5 ft. 4 ins.
Diameter of do.	2 „ 6 „	2 „ 6 „
Number of tubes	7	1
Diameter of do.	2½ ins.	8 ins.
Length of tubes	2 ft. 2 ins.	2 ft. 0 ins.
Area of fire-grate	1¾ sq. ft.	3½ sq. ft.
Height of furnace	1 ft. 9½ ins.	3 ft. 0 ins.
Heating surface	21.2 sq. ft.	26.8 sq. ft.
Ratio of heating surface to grate-area	12.0	8.54
Steam-pressure per square inch	51.5 lbs.	32 lbs.
Coal consumed per hour	36⅔ „	34½ „
Do. do. do. per square foot of grate	21.5 „	9.9 „
Water evaporated per hour from 100° F.	121 „ 2.0 cu. ft.	170 „ 2.74 cu. ft.
Do. do. do. per square foot of grate	1.14 „	0.78 „
Water evaporated per pound of coal	3.3 lbs.	4.9 lbs.

These are results of two days' trial of the first, and one day's trial of the second. The coal used in the first boiler was Parkfield coal, containing 82 per cent of carbon, 5.6 per cent of hydrogen, and 6.1 per cent of oxygen. The coal for the second boiler had about the same composition. The comparative inferiority of the first boiler in efficiency, is, it may be presumed, due to the inferiority of the vertical flue-tube surface. A small vertical boiler, on the premises of Messrs. Wm. Green & Co., London, was tested for evaporation with gas-coke as fuel. The shell is 2 feet 6½ inches in diameter outside, and it stands 5 feet 4½ inches high. The fire-box is 22 inches in diameter, 2 feet 5 inches high above the grate, having 3.33 square feet of grate-area. A 6-inch flue-tube rises through the shell, and there is one 4-inch cross-tube in the upper part of the fire-box. The trial lasted five hours, during which time 87½ pounds of coke was consumed, leaving 6 pounds of ash; and 7.344 cubic feet of water, supplied at a temperature of 70° F., was evaporated, at a pressure of 30 lbs. per square inch. The coke was charged, 10 pounds at a time, at intervals of 33 minutes.

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[1] See Mr. Shaw's paper on "Small Motive Power" in the *Proceedings of the Institution of Civil Engineers*, vol. lxii., 1879–80; page 290.

Coke consumed per hour	17.5	pounds.
Do. do. do. per square foot of fire-grate	5.25	„
Water evaporated per hour	1.47	cubic feet.
Do. do. do. per square foot of grate	.44	„
Do. do. per pound of coke	5.24	pounds.

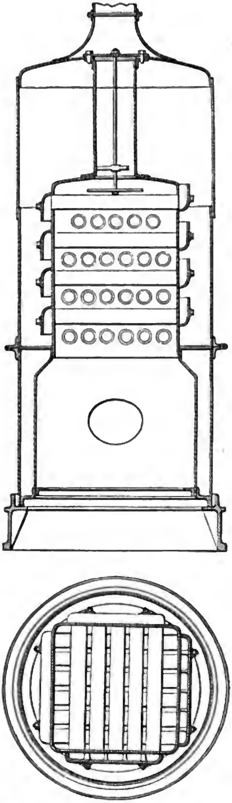
NOZZLE STEAM BOILER, READING IRONWORKS.

The water-pipe or water-tube system has been fully developed by the Reading Ironworks, in their vertical nozzle boiler, figs. 307. The boiler consists of a cylindrical shell, divided transversely into two parts, united by angle-irons and bolts and nuts, which can be separated so that the upper part may be lifted off for examination internally, the upper part of the fire-box being then entirely exposed. The crown of the boiler is convex. The fire-box, at the lower part is cylindrical, and is welded in one piece for boilers of at least 12 horse-power. It is square at the upper part, to receive the transverse water-tubes, which are horizontal, in several rows, crossing laterally and fore-and-aft alternately, and so breaking up the currents of burnt gases as they ascend to the uptake. Nozzles or circulators, of cast iron, are fixed at the ends of each row of tubes, turned downwards at one end, and upwards at the other end of each row. The steam formed in the tubes is, in consequence, discharged from the upturned nozzles, and currents of water flow into the tubes from the downturned nozzles, to supply the place of the steam that is generated and discharged.

Table No. 154.—Vertical Nozzle Boilers: The Reading Ironworks.

Horse-power	2	3	4	5	6	8	10	12	16	20	25	30	40
	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.
Total height of shell above base-plate	5 0	5 4	6 4	6 6½	6 11½	7 4	8 2	8 6	10 0	10 9	12 2	12 0	13 10
Diameter of do. outside	2 0	2 3	2 3	2 8	2 8	3 1½	3 1½	4 0	4 0	4 6	4 6	5 4	6 0
Internal diameter of fire-box	1 5⅝	—	1 8⅝	—	—	2 7	—	—	3 4	—	—	—	5 3
	cwts.	cwts.	cwts.	cwts.	cwts.	cwts.	cwts.	cwts.	cwts.	cwts.	cwts.	cwts.	cwts.
Weight of boiler and chimney only, with circulators	11½	13¾	16½	20	23	31	34	40	49	68	80	102	134
Do. do. including base-plate and fire-bars	14	16¾	19½	25	27	36	40	50	59	86	98	128	163
Weight of mountings	½	½	½	½	½	½	—	—	¼	—	—	—	1¾
Gross weight	14½	17¼	20	25½	27½	36½	—	—	59¼	—	—	—	164¾

The crown of the fire-box is convex, and is connected by the cylindrical uptake to the crown of the shell. The uptake is riveted to the fire-box; but, necessarily, for the sake of the occasional removal of the upper part of the shell, it is fixed to the crown of the shell by means of an angle-iron, and bolts screwed into the angle-iron, which can be unscrewed as required. In order to still further exhaust the heat of the burnt gases, the upward current, as it approaches the uptake, is baffled by a circular iron plate suspended from above over the uppermost row of water-tubes, and a few inches below the opening into the uptake.



Figs. 307.—Vertical Nozzle Steam Boiler, by the Reading Ironworks. Scale 1/24th.

The unit of power is an allowance averaging from 12 to 14 square feet of heating surface per nominal horse-power. Coal of ordinary quality can be burned off at the rate of 4 pounds per nominal horse-power. Water supplied at ordinary temperatures is, it is stated, evaporated at the rate of 10½ pounds per pound of coal. The feed-water is delivered in a single stream into the water-space at a level of about 18 inches above the bottom of the boiler. The working steam-pressure is from 50 lbs. to 60 lbs. per square inch.

The mountings comprise a check feed-valve, a safety-valve, a pressure-gauge, a water-gauge, two test-cocks; wrought-iron fire-door and bracket, four mud-holes, covers, and cross-bars, fire-bars of a specially hard mixture of metal, wrought-iron bearing-bars and studs; and a base-plate fitted with a wrought-iron damper for regulating the supply of air below the grate.

A nozzle boiler of 8 nominal horse-power is illustrated by figs. 307. The shell is 3 feet 1½ inches in diameter outside, and 7 feet 4 inches high to the top of the crown, of ⅜-inch iron plates in two halves, united by 2½-inch angle-irons, and ½-inch bolts and nuts. The dished crown-plate has a rise of 3 inches. The lower part of the fire-box is of ⅜-inch plates, 2 feet 7 inches in diameter inside and 2 feet high, with a 2½-inch water-space all round. It is single-riveted at the joints. It is connected to the shell at the bottom by double angle-iron. The fire doorway is 11 inches by 9 inches, and the joint there is made with a 2-inch-space ring, to meet which the fire-box plate is set out. The fire-box is taken in to receive and be riveted to the upper part containing the water-tubes. The upper part is 2 feet ¾ inch square outside, and 2 feet 9½ inches high to the shoulder, or 2 feet 11¼ inches high to the top of the crown, which has a rise of 1¾ inches. The upper part is crossed by eight rows of water-tubes 2¾ inches in diameter externally, No. 11 Birmingham wire-gauge, or about ⅛ inch, in thickness. There are 6 tubes in each row, except in the two uppermost rows, in each of which there are 5 tubes; making 46 tubes in all. The rows of tubes, as well as the tubes of each row, are 3¾ inches apart between centres, making clear interspaces of 1 inch. The nozzles at each end of each row are in one casting, reaching from side to side, divided by partitions into compartments, one to each tube, so that the circulation for each tube is independent. It is clearly shown by the plan of the tubular section, figs. 307, that the upper chamber of the fire-box is thoroughly occupied by the tubes, and that the current of burnt gases is minutely subdivided in its upward passage to the uptake. The normal water-level is 5 inches above the crown of the fire-box. The uptake and the chimney are 8 inches in diameter inside. The chimney is from 8 feet to 9 feet high above the crown of the shell, or about 15 feet above the level of the fire-grate, which is 3 inches above the bottom of the fire-box, and 2 feet 3 inches below the lowest cross-tubes. The fire-bars are ⅜ inch thick, and the air-spaces are ⅜ inch wide. The safety-valve is 2 inches in diameter.

The area of fire-grate, reckoned for the inside diameter of the fire-box, is 5¼ square feet. The heating surface is about 113 square feet, or 21½ times the area of fire-grate. Two-thirds of this heating surface is supplied by the cross-tubes. Allowing, for ordinary working, 4 pounds of coal per nominal horse-power per hour, the consumption per hour is ($4 \times 8 =$) 32 pounds, which is at the rate of ($32 \div 5.25 =$) 6.10 pounds per square foot of fire-grate per hour. The corresponding quantity of water evaporated is ($32 \times 10.5 =$) 336 pounds, or 5.39 cubic feet per hour, or about 1 cubic foot per square foot of grate-area per hour. At these rates the consumption of water is ($336 \div 8 =$) 42 pounds, or about ⅔ cubic foot per nominal horse-power per hour.

The first nozzle-boiler, constructed of 7 horse-power, was tried in 1872. It was not clad. It had a grate-area of 4 square feet, with 90 square feet of heating surface—about 22½ times the grate-area. Of coal, 37 pounds was consumed per hour, or 9¼ pounds per square foot of fire-grate; and 336 pounds, or 5.39 cubic feet of water supplied at 40° F., was evaporated per hour, or 1.35 cubic feet per square foot of fire-grate; making an evaporation of 9.08 pounds of water per pound of coal; equivalent to 10.90 pounds from and at 212° F.

In the same year, a 5-horse-power nozzle boiler, clothed with felt, was tried, with two different areas of fire-grate. The boiler was 2 feet 9½ inches in diameter, and 6½ feet high. It had 33 iron tubes, 2¾ inches in diameter outside, and 20¾ inches long. The area of fire-grate was 1.3 square feet and 4 square feet; and the heating surface was 71.6 square feet, or respectively 55 times and 18 times the areas of fire-grate. The lowest cross-tubes were 1 foot 11½ inches above the fire-grate. The chimney was

7 inches in diameter. The leading results of three carefully conducted trials are given in the following table, No. 155.

Table No. 155.—RESULTS OF TRIALS OF A 5-HORSE-POWER NOZZLE BOILER. 1872.

No. of Experiment	1	2	3
Area of fire-grate	1.3 sq. ft.	1.3 sq. ft.	4 sq. ft.
Pressure of steam per square inch	48 lbs.	49 lbs.	49 lbs.
Coal consumed per hour	63.4 „	63.4 „	88 „
Do. do. per sq. foot of grate	48.8 „	48.8 „	22 „
Temperature of feed-water	65° F.	65° F.	65° F.
Water evaporated per hour	8.27 cu. ft.	8.52 cu. ft.	11.23 cu. ft.
Do. do. per sq. ft. of grate	6.36 „	6.56 „	2.81 „
Do. per pound of coal	8.12 lbs.	8.37 lbs.	7.95 lbs.
Do. do. do. from and at 212° F.	9.58 „	9.88 „	9.38 „

The results of an experimental trial of a 4-horse-power boiler, by the engineers of the Royal Agricultural Society, at Bedford, in 1874, which lasted 4 hours 2 minutes, show that 80 pounds of coal, and 807 pounds or 12.96 cubic feet of water, were consumed, under a pressure of 20 lbs. per square inch:—being at the rate of 20 pounds of coal and 202 pounds, or 3.24 cubic feet, of water per hour. Taking the area of fire-grate as 2⅓ square feet, 8.60 pounds of coal, and 86.7 pounds or 1.40 cubic feet of water were consumed per hour per square foot of fire-grate. The water was consumed at the rate of 10.09 pounds per pound of coal, equivalent to 11.70 pounds from and at 212° F.

The principal results of tests made at the Reading Ironworks, on a boiler of 40 horse-power on regular duty there, in February, 1876, were as follows. The area of fire-grate was 21.65 square feet.

Table No. 156.—TRIALS OF A 40-HORSE-POWER NOZZLE BOILER. 1876.

Date of Experiment.........1876	Feb. 2.	Feb. 4.	Feb. 7.	Feb. 14.
Duration of experiments	3 h. 22 m.	3 h. 23 m.	2 h. 51 m.	3 h. 3 m.
Pressure of steam per sq. in.	50 lbs.	50 lbs.	20 lbs.	50 lbs.
Coal consumed (small Welsh steam)	504 „	504 „	504 „	476 „
Coal per hour	149.6 „	149.6 „	177.0 „	156.1 „
Do. do. per square foot of fire-grate	6.91 „	6.91 „	8.18 „	7.21 „
Temperature of feed-water	65° F.	65°	62°	104°
Water evaporated	83.24 cu. ft.	83.88 cu. ft.	83.91 cu. ft.	84.23 cu. ft.
Do. do. per hour	24.70 „	24.89 „	29.44 „	27.62 „
Do. do. do. per sq. foot of fire-grate	1.14 „	1.15 „	1.36 „	1.28 „
Water evaporated per pound of coal	10.30 lbs.	10.58 lbs.	10.38 lbs.	10.97 lbs.
Water evaporated per pound of coal from and at 212° F.	12.15 „	12.48 „	12.14 „	12.51 „

The general results of these tests show that at ordinary rates, from 6 pounds to 8 pounds of ordinary coal is consumed, and from 1 cubic foot to 1⅜ cubic feet of water is evaporated, from ordinary temperatures, per square foot of fire-grate per hour; showing an evaporation of from 10 pounds to 10½ pounds of water per pound of good coal, equivalent to from 12 pounds to 12½ pounds from and at 212° F.

The rate of consumption of fuel in proportion to the area of the fire-grate is very moderate, and, in connection with the extensive heating surface presented by the water-pipes, it explains the comparatively low temperatures of the escaping gases claimed for the nozzle boiler.

In the exceptional case of the boiler of 5 horse-power, 11¼ cubic feet of water was evaporated per hour in the third trial; or more than 2 cubic feet per horse-power, at the rate of 9.38 pounds per pound of coal from and at 212° F. The results of the first and second trials, with a reduced fire-grate, show that, whilst the rate of combustion per square foot was more than double that in the third trial with the ordinary grate, the evaporative efficiency also was greater—corroborative of the experimental deductions already made. The results of the third trial, compared with those of the other trials, with lower rates of combustion, also show what has before been remarked, that the more a boiler is forced, and the greater the quantity of water evaporated, the less is the evaporative efficiency.

FIELD BOILER.

The vertical boiler, fig. 308, designed by Mr. Edward Field, is constructed with an ordinary flat-top cylindrical shell, and a plain flat-top fire-box. A large number of lap-welded tubes are fixed into the crown-plate, closed at the lower end, and opening at the upper end into the water-space above the fire-box. These tubes, fig. 309, depend for some distance into the fire-box, and are exposed both to the heat radiated from the fire, and to the heat of the burnt gases which circulate amongst them. Within each tube, another tube, open at both ends, and smaller in diameter, is suspended, so that a rapid circulation of water and steam takes place, by the rising of steam and heated water in the outer tube, and the descent of water relatively less hot, in the internal tube. The external tubes are slightly opened or expanded at their upper ends, and when properly fitted and driven into the tube-plate, or crown-plate of the fire-box, they are kept tightly in place by the pressure of the steam. The upper ends of the inner tubes also are expanded coniformly, to admit of a free entrance for water downwards, and they are suspended by fins or feathers on the outside, so that they do not close the apertures of the fixed or outer tubes, but, on the contrary, leave a free passage upwards and outwards for the mixed currents of steam and water that rise from the outer tubes. By the maintenance of rapid and continuous circulation within the tubes, it results that the transference of heat to the water is augmented, compared with that due to stagnation or to languid circulation.

There is necessarily a cylindrical space in the fire-box, directly under the uptake, which is not occupied by tubes; and it is commanded by a cast-iron baffle-block, suspended a few inches below the entrance to the uptake. By the intervention of this round block, the current of burnt gases is dispersed around it and amongst the circulating tubes. The level of the block, which also acts as a damper, is adjustable by means of a lever from which it is suspended, and a chain which hangs down outside the boiler.

The shells of Field boilers are made either of Landore-Siemens mild steel, or of Yorkshire iron; the fire-boxes and tube-plates and top-plates are of Yorkshire iron. The outside seams of the shell are usually double-riveted, the circular seams single-riveted, and the rivet-holes are drilled. The circulating tubes are ⅛ inch thick for all sizes. The fittings usually supplied are as follows:—Steam stop-valve, safety-valve, pressure-gauge, 2 sets of water-gauges (or 1 water-gauge and 3 gauge cocks), back-pressure valve, and one or two blow-off cocks, anti-priming pipe, a set of fire-bars, and an adjustable baffle-block under the uptake, to regulate the draught.

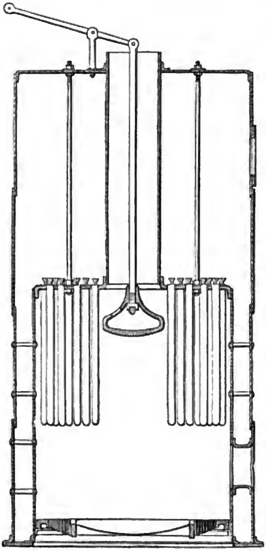


Fig. 308.—Vertical Field Steam Boiler. Scale 1/40th.

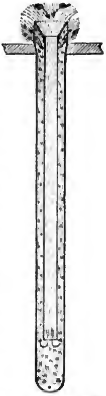


Fig. 309. — Field Boiler: Circulating Water-tubes.

The unit of measure for nominal horse-power is 10 square feet of circulating tube-surface per horse-power, the plate heating surface being additional. With that allowance of surface, it is reckoned that 1 cubic foot of water can be evaporated per horse-power per hour, in ordinary Field boilers. At the same time, it is stated that in boilers for steam fire-engines, the same duty has been performed with an area of 5 square feet.

The following table shows the outer dimensions, weight, and heating surfaces of ordinary vertical Field boilers of from 3 to 60 nominal horse-power:—

Table No. 157.—Field Boilers.

Nominal Horse-power.	Heating Surface.		Height.		Diameter.		Complete Weight for Shipment.	
	Total.	Per Horse-power.					Total.	Per Horse-power.
H.P.	sq. ft.	sq. ft.	ft.	ins.	ft.	ins.	cwts.	cwts.
3	52	17.3	6	0	2	8½	18	6
4	65	16.2	6	6	2	10	22	5.5
5	87	17.4	6	9	2	11½	26	5.2
6	93	15.5	7	0	3	2	29	4.8
8	121	15.1	7	6	3	6	33	4.1
10	145	14.5	7	8	3	9	41	4.1
12	170	14.2	8	2	3	11	43	3.6
14	194	14.0	9	4	4	0	49	3.5
15	218	14.5	9	10	4	2	51	3.4
16	226	14.1	10	3	4	4	57	3.6
18	255	14.2	10	6	4	8	60	3.3
20	282	14.1	10	6	5	0	87	4.4
25	343	13.7	11	0	5	4	110	4.4
30	408	13.6	11	6	5	10	120	4.0
35	462	13.2	11	9	6	0	127	3.6
40	518	13.0	12	0	6	3	138	3.4
50	630	12.6	13	0	6	6	150	3.0
60	740	12.3	14	0	6	9	190	3.2

The ordinary Field boiler is illustrated in fig. 308, by the example of a 50-horse-power boiler, made for the Richmond Water-works. The shell is cylindrical, 13 feet in height, and 6½ feet in diameter externally, of 7/16-inch plates, except the crown-plate and the base-plate, which are ½ inch thick. There is a manhole in the shell near the crown, fortified by a ¾-inch plate-ring riveted to it. The fire-box is cylindrical, about 5 feet 4 inches in diameter inside, and 7 feet 1½ inches high, of 7/16-inch plates, except the crown-plate, which is ⅞ inch thick. The water-space is 6 inches wide. The fire-box is riveted to the base-plate with angle-iron, independently of the shell, and is joined to the shell at the doorway by a plate-iron necking, flanged and riveted to each separately. There is an advantage in this system of necking, as it makes tight joints, and it admits of free circulation of water and steam around it. The fire-box and the shell are connected also by 1-inch screwed stay-bolts at a pitch of 16 inches, and by the uptake, which is of 7/16-inch plate, and is 17 inches in diameter inside, and 1½-inch tie-rods, fixed with double bolts and nuts at each end. The circulating tubes, 250 in number, are 2¼ inches in diameter externally, and 3 feet 10½ inches long. The baffle-block is 19 inches in diameter, and only clears the surrounding tubes by ¾ths of an inch.

The fire-grate is formed of four segmental grids and a central square grating of ordinary fire-bars 3 feet 4 inches in length, supported on an angle-iron ring riveted to the fire-box near the bottom. The surface of the grate is at a level 8 inches above the base-plate or bottom of the boiler, so providing a liberal refuge for the collection of mud below the direct action of the fire. The chimney is 16 feet high.

The boiler of 50 horse-power contains 500 square feet of tube surface, being at the rate of 10 square feet per horse-power; or a total of 630 square feet, being at the rate of 12.6 square feet per horse-power. The area of fire-grate is 22½ square feet, being to the total heating surface in the ratio of 1 to 28.2. The evaporative power, at the assumed rate, is the evaporation of 50 cubic feet of water supplied at ordinary temperatures per hour, or 2.22 cubic feet per square foot of fire-grate. Allowing an evaporative efficiency of 9 pounds of water per pound of coal, the coal would be consumed at the rate of 347 pounds, or 3.1 cwts. per hour; or 15.4 pounds per square foot of fire-grate. The net weight of the 50-horse-power boiler, exclusive of mountings, is 6 tons 3 cwts. 2 qrs. The price of the boiler complete is £380, or £76 per horse-power, or something less than £61, 10*s.* per ton. The working pressure is 60 lbs. per square inch. The ordinary water-level is 18 inches above the fire-box.

This boiler was tried by Mr. Schönheyder for a period of 6 hours 40 minutes, burning Merthyr coal of good quality, though containing a large percentage of slack and dust. The boiler had been cleaned previous to the trial, and was tried on its ordinary pumping work. The quantity of water, supplied at the temperature 49°, evaporated into steam of 55 lbs. pressure per square inch, was at the rate of 1620 pounds, or 26 cubic feet, per hour; or 1.15 cubic feet per square foot of fire-grate. Coal was consumed at the rate of 177.6 pounds, or 1.58 cwts. per hour; or 7.90 pounds per square foot of fire-grate. The water was evaporated at the rate of 9.12 pounds per pound of coal; equivalent to 10.93 pounds from and at 212° F. It was evident that a much larger quantity of steam could have been generated in the same time, had there been a demand for it.

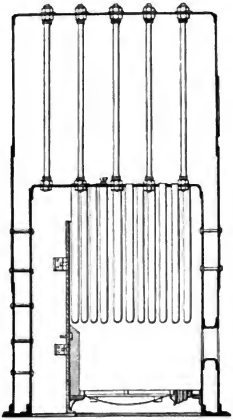


Fig. 310.—Field Boiler, with a Descending Draught. Scale 1/48th.

A modified form of the Field boiler, with a descending draught, is shown in fig. 310. The uptake is dispensed with, and the fire-grate is shortened and inclosed at the back by a vertical partition of cast iron, which rises to a distance of 12 inches below the crown-plate. The whole of the crown-plate within the inclosure is available for the placement of circulating tubes, and the downward draught, regulated by a damper, passes off behind the partition into the flue underground to the chimney.

The boiler here illustrated is of 30 horse-power, and it was constructed for the London Stock Exchange. The shell is 11½ feet high and 5 feet 10 inches in diameter externally, constructed of 7/16-inch steel plates for the cylindrical portion, and 9/16-inch Yorkshire plates for the crown-plate. The fire-box is of 7/16-inch Lowmoor plates and ⅞-inch tube-plate. It is 4 feet 9¾ inches in diameter inside, and 6½ feet high, forming a water-space 5¼ inches wide. It and the shell are constructed and put together similarly to the uptake boiler. The two crown-plates, which are flat, are stayed together with 1¾-inch Lowmoor bolts and double nuts. There are 131 circulating tubes, 2¼ inches in diameter externally, 3 feet 9¾ inches long, presenting together an area of 292 square feet. The partition is 1⅛ inches thick, and is made in three pieces, pinned together. It is cast with a bracket to support the fire-bars and a fire-brick shield or liner 2½ inches thick. The fire-grate is thus of the form of a segment of a circle, 4 feet 9¾ inches in diameter, and 3 feet 6 inches wide. The grate is formed of three lateral segments and a square of fire-bars. The level of the grate is 7½ inches above the bottom of the fire-box, and it is about 2 feet 1 inch below the bottoms of the tubes. The clear height of the crown of the shell, above that of the fire-box, is about 4 feet 9½ inches, of which the water occupies about 13 inches above the crown to the water-level, and the remainder, 44 inches, is steam-space.

The area of the fire-grate is 15.4 square feet, the effect of the compartment being to reduce the area for the complete circle, 18⅓ square feet, by about 3 square feet. The heating surface of the circulating tubes is 292 square feet, representing, say, 30 horse-power. The total surface is 400 square feet, or 26 times the area of grate. The downward-draughtway behind the partition has a sectional area of 2.15 square feet, or 1/14th of the area of grate.

From results of recent tests it appears that, in a downward-draught boiler, similar to the one just described, 10.83 pounds of water was evaporated into steam of 100 lbs. pressure per square inch per pound of coal.

CHAPLIN & CO.'S VERTICAL TUBULAR STEAM BOILERS.

Messrs. Alexander Chaplin & Co., Glasgow, manufacture two forms of vertical boilers—the flue-tube or tubular, and the water-tube or tubulous.

The Vertical Tubular Boiler, of 8 nominal horse-power, is illustrated by fig. 311, in which the distinguishing feature is the contraction or swaging of the flue-tubes a little below the water-line, with the object of retarding the burnt gases as they ascend, and more effectually cooling them; so augmenting the production of steam, and obviating excessive heating of, and consequent injury to, the tubes in the steam-space. It is argued that the burnt gases expand after passing the contractions, and are consequently reduced in temperature. A slight reduction, no doubt, takes place as the result of expansion; but probably the relief is due principally to the diminished lateral pressure of the gases on the surfaces of the upper parts of the tubes, resulting from the sudden enlargement above the contractions.

The boiler is constructed of mild steel. It is 3 feet in diameter outside, for the lower part, and exclusive of the ash-pit, is 7 feet high. The case is made of two rings of plates, of which the upper laps over the lower ring. They are 5/16 inch thick; the tube-plate is ½ inch thick. The fire-box is 29 inches in diameter inside at the level of the fire-grate, and 28 inches at the top; it is 30 inches in total height, and is 22 inches high above the fire-grate. It is made of one plate, ⅜ inch thick, riveted at the joint; the tube-plate is ½ inch thick. The fire-box is set out at the bottom to join the shell, and it forms a water-space 2½ inches wide at the bottom, widening to 3 inches at the top. It is joined to the shell at the doorway with a ring 1⅛ inches thick; and is stayed to it by one line of stay-bolts, 12 inches apart, all round. There are 48 flue-tubes, 2¼ inches in diameter, ⅛ inch thick, and having a clear length of 4 feet 5½ inches between the tube-plates. They are slightly inclined towards the centre upwards, for convenience in clearing the conical uptake on the top of the boiler. They are swaged to a diameter of 1½ inches externally, and about 1¼ inches internally: the normal flue-area being reduced at the swaging to two-fifths. The tubes are fixed into the two crown-plates by expanding, and are turned over or headed on the plates. The water-level is 4 inches above the swaging of the tubes, or about 2½ feet above the fire-box, leaving a steam-space about 2 feet high. The volume of water is 15.3 cubic feet, and that of the steam-space is 10.3 cubic feet. The chimney is 8 inches in diameter inside, and No. 10 wire-gauge in thickness; it is 10¾ feet high above the sole of the boiler, or 10 feet 1 inch above the level of the fire-grate. The fire-grate is formed with two lateral cast-iron segments, and parallel cast-iron bars between them; ⅝ inch thick with ⅜-inch air-spaces. The circular seams are single-riveted; the vertical seams are double-riveted, by machine. The rivets are ¾ inch in diameter, at 2 inches of pitch for single-riveting, with 2¼ inches of lap; for double-riveting, the lap is 3 inches, and the angular pitch is 2 inches. The feed-water is delivered direct from the check-valve near the bottom of the boiler. The ash-pit may be constructed of a few courses of brick, in horse-shoe form. The working pres-

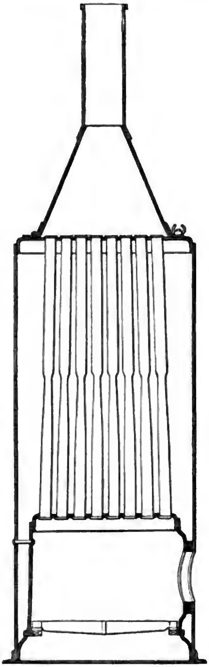


Fig. 311.—Vertical Steam Boiler, by Messrs. A. Chaplin & Co. Scale 1/24th.

sure is 75 lbs. per square inch, and boilers are tested by cold water to double that pressure.

The area of the fire-grate is 4.59 square feet. The wet heating surface consists of 14 square feet of plate surface, and 70 square feet of tube surface; together, 84 square feet, or 18.3 times the grate-area. There is 56 square feet of steam heating surface.

The unit of nominal horse-power is the evaporation of 1 cubic foot of water per hour. In the 8-horse-power boiler there is (84 ÷ 8 =) 10½ square feet of wet heating surface per horse-power; and (4.59 ÷ 8 =) .57 square foot of fire-grate per horse-power. Correspondingly, the rate of evaportion is 500 pounds, or 8 cubic feet of water from ordinary temperatures, being at the rate of 1.74 cubic feet per square foot of fire-grate. It is stated that 62½ pounds of coal are consumed per hour, or 13.6 pounds per square foot of fire-grate; evaporating 8 pounds of water per pound of coal.

The weight of the boiler is 28 cwts., not including that of boiler mountings; or 3½ cwts. per horse-power. The price is £55, or about £7 per horse-power; or about £40 per ton, without boiler mountings.

Particulars of vertical tubular boilers, of from 4 to 10 nominal horse-power, are given in the tablet following:—

Chaplin's Vertical Tubular Boilers.

Nominal Horse-power		4	6	8	10
Height, exclusive of ash-pit	ft. ins.	7 0	7 0	7 0	7 6
Diameter	„ „	2 6	2 9	3 0	3 3
Heating surface—plate	sq. ft.	11	14	14	20
Do. wet tube-surface	„	35	52	70	82
Do. sum	„	46	66	84	102
Water capacity, ...	cu. ft.	9.4	12.7	15.3	18.5
Steam-space	„	8.0	9.1	10.3	11.7
Weight, with fire-bars, doors, and chimney...	cwts.	20	23	28	33
Price, with do. do. do. ...	£	40	47	55	64

Chaplin's Vertical Water-tube Boiler, of 7 nominal horse-power, is illustrated by fig. 312. The special features are the dependent water-tubes, which hang from the crown-plate of the fire-box. The crown-plate being dished, and the tubes screwed into it at right angles, the tubes are inclined radially towards the centre-line of the boiler, and are in a position favourable for the circulation of the burning gases between and around the tubes, and the circulation of water and steam within them.

The boiler is constructed of mild steel. The shell is 7 feet 2 inches high, exclusive of the ash-pit, which may be of brickwork, and 3 feet 1 inch in diameter; and is of $^{5}/_{16}$-inch plates. The walls of the fire-box are of ⅜-inch plates, 31 inches in diameter inside at the level of the fire-grate. The crown-plate is ⅝ inch thick. The fire-box has a total height of 4 feet 8 inches; and is 4 feet 1 inch above the level of the fire-grate. There is a cluster of 24 hanging tubes in the fire-box, 2¼ inches in diameter externally, and ⅛ inch thick; and 2½ feet long, below the tube-plate;

hanging about 18½ inches clear of the fire-grate. The fire-box is set out at the base, to join the shell, and is joined with a ring at the doorway. The water-space is about 2¼ inches wide at the bottom, and widens upwards to 3 inches at the top. There is a clear height of about 2 feet 4½ inches from the crown of the fire-box to that of the shell, in which the water-level is 6 inches above the fire-box, leaving 1 foot 10½ inches for steam-space. The uptake is conical, 5 inches in diameter at the crown of the fire-box, expanded to 8 inches at the crown of the shell, to meet the chimney.

The heating surface comprises 33 square feet of plate-surface, and 37 square feet of tube-surface; together 70 square feet, or 10 square feet per horse-power. The area of fire-grate is 5¼ square feet, and is to the heating surface as 1 to 13.3. Evaporating 1 cubic foot per horse-power, or 7 cubic feet per hour, 1.33 cubic feet is evaporated per square foot of fire-grate per hour. Allowing 8 lbs. of water evaporated per pound of coal, there is a consumption of 54.6 pounds of coal per hour, or 10.4 pounds per square foot of fire-grate. The boiler weighs 25 cwts., exclusive of the weight of the boiler mountings, or 3.6 cwts. per horse-power. The price is £52, or £7, 9*s*. per horse-power; or £41, 12*s*. per ton of net weight.

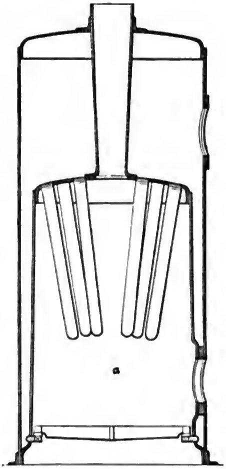


Fig. 312.—Vertical Water-tube Boiler, by Messrs. A. Chaplin & Co. Scale 1/24th.

Chaplin's Vertical Water-tube Boilers.

Nominal Horse-power		5	7	9	11
Height, exclusive of ash-pit	ft. in.	6 10	7 2	7 6	7 10
Diameter	„ „	2 10	3 1	3 4	3 7
Heating surface—plate	sq. ft.	28	33	40	46
Do. wet tube-surface	„	24	37	50	67
Do. sum	„	52	70	90	113
Water capacity	cu. ft.	6.0	7.9	9.0	11.5
Steam-space	„	10.5	14.0	15.0	17.7
Weight, with fire-bars, doors, and chimney	cwts.	23	25	30	36
Price, with do. do. do.	£	44	52	62	74

COCHRAN & CO.'S VERTICAL TUBULAR STEAM BOILER.

The vertical boiler, figs. 313 and 314, manufactured by Messrs. Cochran & Co., Birkenhead, is constructed with a dome-shaped fire-box, and a group of horizontal flue-tubes above the fire-box, encased in a vertical cylindrical shell. The burning gases pass from the fire-box by a circular opening, or flue-pipe, at the back, into a combustion-chamber, from which they pass through the flue-tubes to the front into the smoke-box and thence into the chimney. By the domical form of the fire-box, the surface is well disposed for receiving the radiant heat from the fire, and spacious waterway is provided externally to it, for the free circulation of water and escape of steam. There is also wide water-space at each side of the group of flue-tubes. In the hemispherical or the segmental top of the boiler considerable steam-space is provided, whilst it stands high for taking off the steam and obviating the chance of priming. The combustion-chamber is closed at the back by an iron plate lined with fire-brick; and it is obvious that the fire-brick must be highly heated by the impinging flames. The heating surface within reach receives the benefit of the heat radiated by the fire-brick.

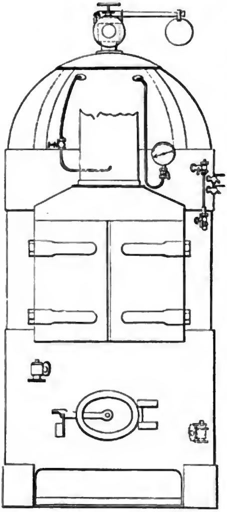
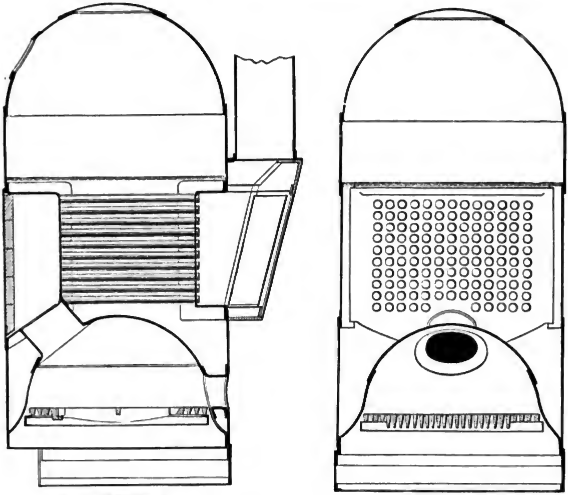


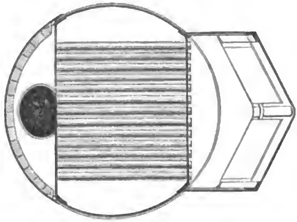
Fig. 313.—Vertical Steam Boiler, by Messrs. Cochran & Co., Birkenhead. Scale about 1/38th.

The boilers are constructed entirely of steel, and according to the regulations of the Board of Trade. The fire-box is made with a circular crown-piece and three lateral plates, shaped and riveted together; except in the six smallest sizes, for which the fire-box is made with two lateral plates. The flue-pipe is one piece, welded or, as in most cases, stamped out of a solid plate; flanged to join the fire-box. The tube-plates are flanged to join the shell. The shell is constructed of three tiers of plates, with the crown, which is constructed of five plates—a central crown-piece, and four lateral segments. The fire-box is set out at the lower edge to join and be riveted direct to the shell. It is also connected to the shell at the doorway by a flanged ring. The flue-tubes are fixed in place by means of expanders. They are beaded or turned over on the back tube-plate; and they project about ¾ inch beyond the front tube-plate, in order that, if necessary, they may afterwards be driven back and beaded anew. Several flue-tubes are utilized as stays, fig. 315, by being screwed into the back tube-plate with one nut on the outside. At the front end they are fastened and made tight with a nut at each side of the tube-plate. The fire-grate is of cast-iron—consisting of four grated segments and a square of fire-bars, and occupying the whole horizontal area of the fire-box. The fire-bars are 1¼ inches wide at the upper edge, and ½ inch at the lower edge, with ⅝-inch air-spaces.

The mountings consist of a water-gauge, 2 test-cocks, a lever safety-valve (figs. 316), a stop-valve, a feed check-valve, a pressure-gauge, a steam-jet pipe and cock, and a blow-off cock.

The Cochran boilers are not rated by nominal horse-power. The unit of power or capacity is the evaporation of 1 cubic foot of water by 10 square feet of heating surface. At the same time, it is assumed that this is also the unit of horse-power. This type of boiler is constructed with from 30 square feet to 800 square feet of heating surface, reputed to be capable of evaporating from 3 cubic feet to 80 cubic feet of water per hour, with from 2¾ square feet to 28 square feet of grate-area.





Figs. 314.—Cochran's Boiler: Sectional Views. Scale about 1/38th.

The thicknesses of steel plates used in the construction of a few of the Cochran boilers, of different sizes, are given in the following tablet for a working pressure of 80 lbs. per square inch. The boilers are constructed for pressures of from 60 lbs. to 100 lbs. per square inch. For iron plates, when ordered, the thicknesses are greater than for steel plates.

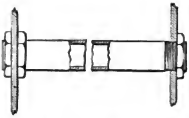
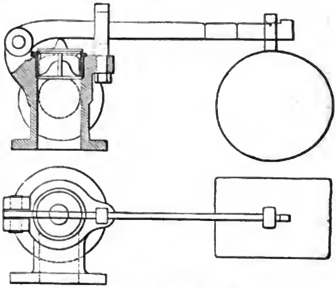


Fig. 315.—Cochran Boiler: Stay-tubes. Scale 1/10th.



Figs. 316.—Cochran Boiler: Safety-valve. Scale 1/10th.

Cochran Boiler.—Thickness of Steel Plates, for a working pressure of 80 *lbs. per square inch.*

Heating Surface.	Diameter of Shell.		Height, including Ash-pit.		Shell.			Fire-box, &c.				Back-plate of Combustion Chamber.
					Top and Bottom Plates.	Middle Plates.	Crown Plates.	Crown Plates.	Side Plates.	Tube Plates.	Flue-pipe and Fire-hole Pipe.	
square feet.	ft.	in.	ft.	in.	inch.	inch.	inch.	inch.	inch.	inch.	inch.	inch.
30	2	3	5	3	7/32	5/16	9/32	¼	7/32	11/32	⅜	3/16
60	3	0	6	9	9/32	⅜	11/32	5/16	9/32	11/32	⅜	3/16
120	4	0	9	0	11/32	15/32	13/32	⅜	⅜	7/16	⅜	¼
220	5	0	10	9	⅜	½	11/32	⅜	15/32	19/32	½	5/16
400	6	0	14	0	13/32	17/32	⅜	7/16	9/16	9/16	½	5/16

The Cochran Boiler.—Flue-tubes, for a working pressure of 80 *lbs. per square inch.*

Heating Surface.	Number and Description of Flue-tubes.	Diameter Outside.	Length between Plates.		Thickness of Stay-tubes.	Thickness of other Tubes.
		inches.	ft.	in.	inch.	B.W.G.
30	30 plain 2 stay } 32	1½	1	4¼	¼	No. 11
60	49 plain 5 stay } 54	1¾	1	8½	¼	No. 11
120	52 plain 8 stay } 60	2	2	7½	¼	No. 11
220	70 plain 16 stay } 86	2½	3	2	5/16	No. 11
400	120 plain 20 stay } 140	2½	3	8	5/16	No. 11

The horizontal or circular lap seams are single-riveted, and the longitudinal or vertical seams are double-riveted, by machine. The following are the diameters and pitches of rivets for steel plates of different thicknesses.

Cochran Boilers.—Riveting.

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets.		Zigzag Pitch, Double-riveting.	Lap.	
		Single-riveting.	Double-riveting.		Single.	Double.
inch.	inch.	inches.	inches.	inches.	inches.	inches.
$\frac{3}{8}$	$\frac{11}{16}$	2	$2\frac{5}{16}$	2	$2\frac{1}{16}$	$3\frac{13}{16}$
$\frac{13}{32}$	$\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{1}{2}$	$2\frac{1}{8}$	$2\frac{1}{4}$	$4\frac{1}{8}$
$\frac{7}{16}$	$\frac{13}{16}$	$2\frac{1}{4}$	$2\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{7}{16}$	$4\frac{1}{2}$
$\frac{15}{32}$	$\frac{7}{8}$	$2\frac{3}{8}$	3	$2\frac{3}{8}$	$2\frac{5}{8}$	$4\frac{13}{16}$
$\frac{1}{2}$	$\frac{15}{16}$	$2\frac{1}{2}$	$3\frac{3}{16}$	$2\frac{1}{2}$	$2\frac{13}{16}$	$5\frac{3}{16}$

The approximate weights and prices of Cochran boilers are as follows:—

Cochran Boilers.—Approximate Weights.

Diameter of Boiler.		Heating Surface.	Boiler.		Smoke-box.		Fire-bars.		Fire-door and Mud-lid.		Mountings.		Chimney.		Total Weight.		Price of Boilers, net.	Price Complete.[1]
ft.	ins.	sq. ft.	cwts.	lbs.	cwts.	lbs.	cwts.	lbs.	cwts.	lbs.	cwts.	lbs.	cwts.	lbs.	cwts.	lbs.	£	£
2	3	30	9	2	0	77	0	84	0	33	0	70	0	70	12	54	$26\frac{1}{2}$	36
3	0	60	17	56	1	28	1	28	0	49	0	77	0	98	22	0	52	69
4	0	120	34	0	2	0	3	56	0	70	1	0	1	14	42	28	95	120
5	0	220	60	0	3	0	5	84	0	77	2	28	1	28	72	105	160	195
6	0	400	90	0	5	0	9	0	0	77	3	0	2	0	109	77	235	287

[1] *Note.*—The prices provide for 6 feet of length of chimney.

Assuming the nominal horse-power of the boilers at the rate of 10 square feet of heating surface per horse-power, the following proportions of weight, price, and horse-powers are obtained:—

Cochran Boilers.—Relative Weight, Price, and Horse-power.

Diameter of Boiler.		Heating Surface.	Nominal Horse-power.	Weight per Horse-power.				Price Complete per Horse-power.	Price per Ton of Boiler, net.	Price per Ton Complete.
				Boiler, net.		Complete.				
ft.	ins.	sq. ft.	H.P.	cwts.	lbs.	cwts.	lbs.	£	£	£
2	3	30	3	3	19	4	18	12	56	76
3	0	60	6	2	103	3	75	$11\frac{1}{2}$	$59\frac{1}{2}$	79
4	0	120	12	2	93	3	57	10	56	71
5	0	220	22	2	81	3	34	9	53	65
6	0	400	40	2	14	2	69	7.2	55	68

The boiler illustrated by figs. 313 and 314, has 350 square feet of heating surface, equivalent to 35 nominal horse-power, reckoned at the rate of 10 square feet of surface per horse-power. The heating surface consists of—

Plate surface..................	54 square feet, or	15.4 per cent.
Tube surface..................	296 „	or 84.6 „
Total.........	350 „	100 „

The area of the fire-grate is 18½ square feet, and the ratio of the grate-area to the heating surface is as 1 to 19. The shell is 5¾ feet in diameter, and the height is 12¼ feet net, or 13 feet above the bottom of the ash-pit. The fire-box is 4 feet 9½ inches in diameter at the level of the grate, and is 2 feet 5 inches high above the grate, forming a hemisphere. The level of the grate is 12½ inches above the lower edge of the fire-box. The water-space is 5 inches wide at the level of the grate, and by the form of the fire-box, quickly widens upwards. The tube-plates are 3 feet 7¼ inches apart, measured over the outer faces of the plates, or about 3½ feet between the plates. There are 108 plain flue-tubes, and 19 stay-tubes; together, 127 tubes, 2½ inches in diameter outside. The stay-tubes are fastened in the manner already described, as shown in fig. 315. The tubes are placed at about 3½ inches apart between centres, allowing 1 inch clear space between the tubes. The fire-hole is 10½ inches by 18 inches, the lower side being level with the fire-grate. The flue-pipe leading to the combustion-chamber is 15½ inches in diameter, and the combustion-chamber is 15 inches long between the fire-brick lining and the tube-plate. The fire-brick lining is 3 inches thick, and is fastened to the back plate of the chamber. The back plate is fixed to the boiler by six or eight lugs, tightened up by screwed bolts. By slacking the bolts, the back plate can be lifted off by means of two rings fixed on the plate. The normal water-level in the boiler is 6 inches above the uppermost flue-tubes, and leaves a steam-space 4 feet 7 inches high to the top of the boiler. The chimney is 19½ inches in diameter.

The thicknesses of plates and tubes for pressures of from 60 lbs. to 100 lbs. per square inch, in the 35-horse-power boiler, are given in the following tablet:—

Cochran Boiler, 35 *horse-power:—Thickness of Plates and Tubes for various Pressures.*

Effective Pressures per Square Inch........	60 lbs.	70 lbs.	80 lbs.	90 lbs.	100 lbs.
	inch.	inch.	inch.	inch.	inch.
Shell: top and bottom plates..........	⅜	⅜	13/32	7/16	½
Do. middle plates.....................	½	½	17/32	9/16	21/32
Do. crown plates	11/32	11/32	⅜	13/32	7/16
Fire-box: crown plate	⅜	⅜	7/16	15/32	½
Do. side plates....................	7/16	½	½	15/32	½
Do. fire-hole pipe.................	7/16	7/16	½	½	9/16
Flue-pipe	7/16	7/16	½	½	9/16
Tube-plates...............................	7/16	½	19/32	⅝	11/16
Combustion-chamber, back plate.....	5/16	5/16	5/16	5/16	5/16
Flue-tubes, thickness....................	¼	¼	5/16	5/16	5/16
Ash-pit....................................	⅜	⅜	⅜	⅜	⅜

The sectional area of flueway through the tubes is 2.43 square feet, or it is to the grate-area in the ratio of 1 to 7.6. The chimney has 2.10 square feet of sectional area, or $^1/_9$th of the grate-area.

The safety-valve, figs. 316, page 746, is mounted on the crown of the boiler, and the steam stop-valve, on Mather & Platt's system, fig. 317, is bolted to it. These are of cast iron with gun-metal fittings. The thoroughfare for steam is 3½ inches in diameter, with ⅝ inch thickness of metal for the safety-valve, and $^9/_{16}$ inch for the stop-valve. The safety-valve is 3½ inches in diameter; the fulcrum of the lever is 3¼ inches from the centre of the valve, and the lever has 21½ inches of clear length, from the fulcrum to the last notch, in which the weight, 115 pounds, is hung for a maximum pressure of 80 lbs. per square inch. The face of the valve is coned to the angle 45°, and rests on the square edge of the seat. The steam stop-valve is 3½ inches in diameter, and is similarly formed. It is raised and lowered by means of a screw.

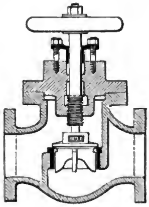


Fig. 317.—Cochran Boiler: Steam Stop-valve. Scale 1/10.

For boilers adapted for service on ships, a pair of spring safety-valves, like the usual type of marine spring safety-valves, is placed on the crown of the boiler, as illustrated by fig. 318, for the 5 feet 9 inch boiler. Each spring is a coil of steel, $^{33}/_{64}$ inch, or .517 inch square in section, in 14 coils, of a mean diameter of 2½ inches. The spring is seated on a sleeve lodged on the valve-spindle, and is covered by a cap which takes the pressure of an adjustable screw for producing the requisite degree of compression—equivalent, in this case, to 80 lbs. per square inch on the safety-valve. The safety-valve is 3 inches in diameter, bevelled to the angle 45°, and seated on a square edge. The spring is encased in a cast-iron cylinder, and the screw, after having been adjusted, is gagged by a ring inserted under its head, and covered by a kind of inverted cup to prevent tampering. The valve can be turned on its seat, when required, by the cross handle cottered on the upper end of the spindle. The two springs with their casings are fixed on a cast-iron base holding the two safety-valves, joined with a pipe-neck, 4¾ inches in diameter, to the boiler. When the pressure in the boiler rises to the maximum, the valve lifts and discharges steam at such a rate that the pressure in the boiler sinks. The valve usually closes, and stops further escape of steam, when the pressure has fallen 1 lb. per square inch below the maximum for which the spring is adjusted. A

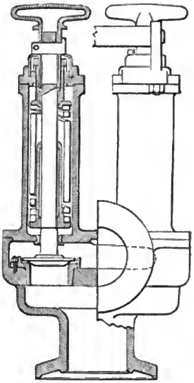


Fig. 318.—Cochran Boiler: Spring Safety-valves. Scale 1/10th.

round shaft is carried in two standards fixed on the top of the casing, as a fulcrum for the lever used for raising the valves when required for allowing steam to escape, or to verify the state of the valves.

A separate steam stop-valve, fig. 319, 3 inches in diameter, is fixed near the spring safety-valves. A segmental piece of pipe is laid inside the shell, blocked at the inner end, and perforated on the upper side by saw-cuts, for collecting and conducting steam to the stop-valve.

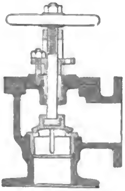


Fig. 319.—Cochran Boiler: Steam Stop-valve, in connection with the Spring Safety-valves.

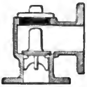


Fig. 320.—Cochran Boiler: Feed Check-valve. Scale 1/10th.

The feed check-valve, fig. 320, is similar in construction to the other valves just described. The valve and the feed-pipe are 2 inches in diameter.

The approximate weights of the boiler of 35 horse-power are as follows. To these are added the weights of similar boilers of 60 and 80 nominal horse-power for comparison:—

Weight of Cochran Boilers.

Nominal horse-power	35		60		80	
Heating surface . . . sq. ft.	350		600		800	
Diameter . . . ft. ins.	5	9	7	0	8	0
	cwts.	lbs.	cwts.	lbs.	cwts.	lbs.
Boiler alone	85	0	125	84	158	56
Smoke-box	4	0	9	0	10	0
Fire-bars	8	0	14	56	19	28
Fire-door and mud-lids	0	77	0	84	0	84
Mountings	2	56	2	84	3	0
Chimney (6 feet)	1	61	2	0	2	14
Total weight	101	82	154	84	193	70

The total weight is here shown to be upwards of 5 tons. The price is £215, exclusive of fire-bars, the price of which is £7, 15*s.*, and mountings £42, 10*s.*, making a total of £265, 5*s.*, or about £53 per ton, or £7, 11*s.* 5*d.* per nominal horse-power of 10 square feet of heating surface.

Trials.—A Cochran boiler of 6 nominal horse-power was tested by Mr. J. H. Harrison in 1879. It was 2 feet 9 inches in diameter, and 5¼ feet high; with a normal grate-area of 4.82 square feet, reduced, for the trial, to 2.5 square feet, and 63 square feet of heating surface—25 times the grate-area. From the printed report it is gathered that 43.8 pounds of Nixon's Navigation steam coal was consumed per hour, or 17½ pounds per square foot of fire-grate per hour. To attain this rate of combustion the draught was urged by a steam-jet in the chimney. Six cubic feet, or 374½ pounds of water, delivered to the boiler, at the temperature 46.5° F., was evaporated per hour, being at the rate of 2.40 cubic feet per square foot of grate-area per hour. Per pound of coal, 8.54 lbs. of water was evaporated, equivalent, it is stated, to 10.195 pounds per pound of coal, from and at 212° F. The temperature of the gases in the chimney averaged 375° F.; the maximum temperature being 425° F.

The ordinary shop-boiler at the factory of Messrs. Cochran & Co. was tested, in 1881, by the author. It is 6 feet in diameter, having a total height of 13 feet. The fire-grate is 5 feet 5 inches in diameter, having 23 square feet of area; but it was partially blocked by brickwork of a segmental form, and reduced to a working area of 15 square feet. There is 60 square feet of plate heating surface, and 265 square feet of tube-surface—together 325 square feet, which is 21½ times the free area of fire-grate. The boiler was tested under the ordinary working conditions of the factory. Two qualities of fuel were used—a common kind of slack, and Tyler's South Wales steam coal.

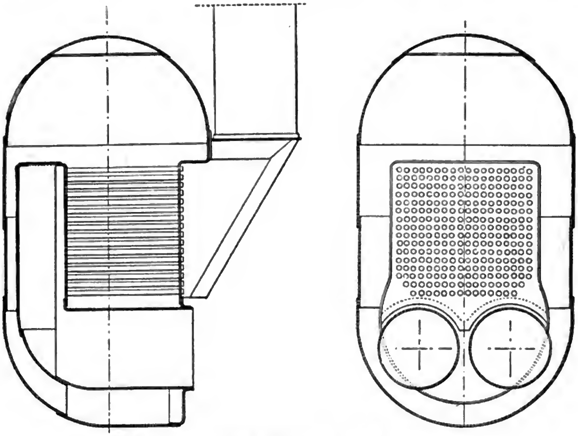
Fuel used	SLACK.	WELSH COAL.
Pressure of steam on gauge, per square inch	80 lbs.	80 lbs.
Temperature of feed-water, average	170° F.	165° F.
Water evaporated per hour	24 cu. ft.	22.5 cu. ft.
Do. do. per square foot of grate	1.6 cu. ft.	1.5 cu. ft.
Fuel consumed per hour	245 lbs.	162½ lbs.
Do. do. per square foot of grate	16.33 lbs.	10.83 lbs.
Water actually evaporated per lb. of fuel	5.94 lbs.	8.42 lbs.
Equivalent water evaporated from and at 212° F.	6.41 lbs.	9.13 lbs.
Average temperature of the gases in the chimney	545° F.	
Maximum do. do. do.	560° F.	

Messrs. Newall & Ogilvie, Bristol, tested a 25-horse-power Cochran boiler, in ordinary condition, at their works, in 1881. The boiler was 5¼ feet in diameter, and 11¼ feet high; having 16 square feet of grate-area, and 250 square feet of heating surface—15.6 times the grate-area. Good Welsh slack was used, at the rate of 157½ pounds per hour, or 9.84 pounds per square foot of grate-area. Steam of 51 lbs. pressure per square inch was generated, from feed-water supplied at 50° F., evaporated at the rate of 21.03 cubic feet per hour, or 1.31 cubic feet per square foot of grate; being at the rate of 8.33 pounds per pound of fuel, or 9.91 pounds from and at 212° F.

Tests made under such different conditions as those here mentioned, are not readily comparable. Mr. Harrison's test shows what may be done occasionally, for a short period—by very much contracting the fire-grate and intensifying the combustion. But it also shows that 1 cubic foot of water could be evaporated per hour, by about 10 square feet of heating surface; and what was done with a contracted grate could, of course, be done with a larger grate, though with less evaporative efficiency.

COCHRAN & CO.'S VERTICAL TUBULAR STEAM BOILER, WITH HORIZONTAL FURNACES.

A compact adaptation of the ordinary furnace-tube to the tubular speciality of the Cochran boiler, is illustrated by figs. 321, in which the domical fire-box is replaced by two furnace-tubes. The shell is 15 feet high, and 8 feet in diameter; but the lower part is flat-fronted to join the furnace-tubes. These tubes are 3 feet in diameter internally, and about 4¾ feet long. The flue-tubes are grouped together horizontally between two tube-plates, which are 4 feet 6½ inches apart outside to outside. The furnace-tubes are flanged to take the tube-plate behind and stay it. The back of the combustion-chamber is formed solid with the boiler, as a water-space, and is stayed with bolts at 8½-inch centres, screwed and nutted at both ends. There are 282 flue-tubes, 2½ inches in diameter outside, No. 11 Birmingham gauge in thickness, except the stay-tubes, 5/16 inch thick; at a pitch of about 3⅝ inches horizontally, and 3¾ inches vertically. The fire-grates are 4¾ feet long. The fire-bars are 1⅛ inches thick at the top, and ⅝ inch at the bottom, with ⅝-inch air-spaces. The boiler is proved with a pressure of 150 lbs. per square inch; the working pressure is 75 lbs. per square inch.



Figs. 321.—Cochran Boiler, with Horizontal Furnaces. Scale 1/58th.

There is 28½ square feet of grate-area. Of heating surface, there is 125.6 square feet of plate-surface, and 824.4 square feet of tube-surface; together 950 square feet, or 33.3 times the grate-area.

THE SHARPE-PALMER VERTICAL TUBULAR BOILER.

The Sharpe-Palmer boiler, fig. 322, is a vertical cylindrical boiler, containing a high fire-box, in the upper part of which a group of horizontal flue-tubes is placed, within a cylindrical barrel projected from one side of the fire-box, reaching nearly to the opposite side. The barrel has free communication with the water-space, and means of additional circulation are provided by a pipe which connects the lower part of the water-space with the under side of the barrel, and another pipe, placed centrally, from the upper side of the barrel to the crown of the fire-box. A loose cap is dropped into place over the central pipe, to disperse the current of water and steam that rises from it. The crown-plates of the shell and the fire-box are both dished; and they are stayed together by 1¼-inch rods, with washers and double nuts at each end. The chimney is placed on the smoke-box fixed on the outside of the shell.

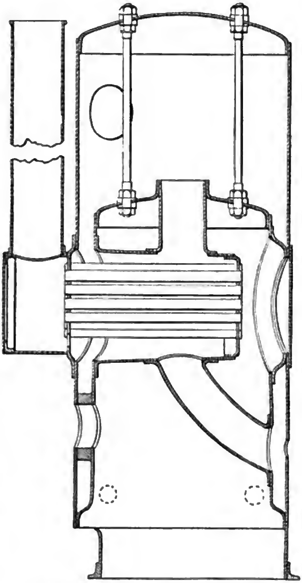


Fig. 322.—The Sharpe-Palmer Vertical Tubular Steam Boiler. Scale 1/24th.

The boiler, illustrated by fig. 322, is one of 8 nominal horse-power. The total height of the shell from the floor to the spring of the crown is 8 feet 10 inches; or, omitting the ash-pit, 8 feet; and the diameter is 3 feet 6 inches. The fire-box is about 35 inches in diameter at the fire-grate, giving 6.7 square feet of grate, or .84 square foot per horse-power; and it has a total height of 5 feet 5 inches; or 4 feet 10½ inches above the fire-grate. The barrel is about 2 feet 2 inches clear above the fire-grate; and it stands about 10½ inches clear below the crown-plate at the centre. The burning gases are therefore free to circulate around and above the barrel before entering the flue-tubes. From these they are discharged into the smoke-box, and to the chimney. There are 28 flue-tubes in the barrel, 2 inches in diameter externally, and 2 feet 8 inches long between plates, at a pitch of 2¾ inches. The heating surface is 100 square feet, which is 15 times the grate-area, and is at the rate of 12½ square feet per horse-power.

The shell-plates and fire-box plates are of ⅜-inch iron plates; and the crowns are of 7/16-inch mild steel. The boiler is proved to a pressure of 150 lbs. per square inch; and it weighs, without mounting, 29 cwts. The price of the boiler, net, is £66, or £8, 5*s*. per horse-power, or £45, 10*s*. per ton of net weight. The total price, including the ash-pit, fire-bars, and mountings, is £79, 15*s*.; or about £10 per horse-power.

A Sharpe-Palmer boiler of 12 horse-power was tested at Newark, in May, 1883, by Mr. Wilson Hartnell. The shell is 4 feet in diameter, and 9 feet high above the ash-pit, or about 10 feet high, including the ash-pit. The fire-box is 3 feet 5½ inches in diameter, and is 6 feet in total height, or 5½ feet high above the fire-grate. There are 34 flue-tubes 2 inches in diameter and 3 feet 2 inches long. The chimney is about 12¾ inches in diameter. The area of fire-grate is 9 square feet, or ¾ square foot per horse-power. The heating surface is 141 square feet, or 15⅔ times the grate-area, and is at the rate of 11¾ square feet per horse-power. The flueway through the tubes is .56 square foot, or 1/16th of the grate-area; the sectional area of the chimney is .90 square foot, or 1/10th of the grate-area.

The coal used was Shireoaks,—a highly-bituminous free-burning coal, leaving but little ash. The fire-door was kept partly open to prevent the smoke. Steam of from 65 lbs. to 68 lbs. pressure per square inch was maintained during the trial. Steam having been got up to a pressure of 60 lbs., the fire was raked out, and the ash-pit cleared. The fire was relighted with 10 pounds of dry wood, and to maintain the draught a steam-jet in the chimney was turned on. The trial lasted 2½ hours, during which time 317 pounds of coal was consumed; and 255 gallons, or 2551.6 pounds, or 40.89 cubic feet of water supplied at 54° F. was evaporated.

Coal consumed per hour	126.8 pounds.
Do. do. per square foot of grate	14.1 „
Ash, 8 pounds, or	2½ per cent.
Water at 54° F. consumed per hour	16.36 cubic feet.
Do. do. do. per square foot of grate	1.82 „
Do. do. per pound of coal	8.05 pounds.
Do. do. do. from and at 212° F.	9.50 „

CHAPTER XIV.—SECTIONAL STEAM BOILERS.

It is a truism that the larger the diameter of a steam boiler the greater is the bursting stress under a given pressure per square inch. It is also a truism that the smaller the parts of a boiler, the more easily transportable it is. In early days, no doubt, and even in more recent days, heavy boilers of large dimensions were, comparatively speaking, dangerous; and subdivided boilers, in parts of small dimensions, have from early times been proposed and put in practice for the generation of steam of high pressure. The earliest subdivided boiler,—or sectional boiler, as such boilers are called,—appears to have been designed by John Blakey about the year 1774, represented by fig. 323, "to generate steam of the required elasticity, and in sufficient quantity."[1] It consists of three water-pipes, inclined alternately, connected at the ends by bent tubes; so that the steam formed in the boiler rose to the upper part, to supply the engine.

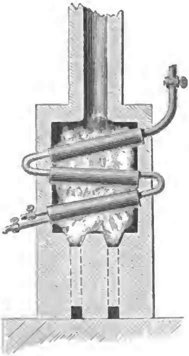
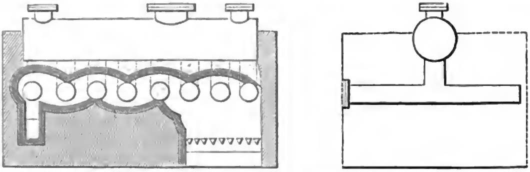


Fig. 323.—Sectional Boiler, by John Blakey, 1774.

Arthur Woolf, in 1804, patented his sectional boiler, figs. 324. A number of cast-iron water-pipes are placed horizontally in a row, connected by branch pipes to a pipe of larger diameter, or receiver, placed above them. The pipes are full of water, which reaches half-way up into the receiver, the upper half of the receiver being space for steam. The pipes are laid transversely in the furnace, and are traversed by the burning gases, which pass over and under them alternately.



Figs. 324.—Sectional Steam Boiler, by Arthur Woolf, 1804.

Mr. Goldsworthy Gurney, in 1825, patented a steam-carriage with a water-tube, tubulous, or sectional boiler. After experimenting with various forms of boiler, the last arrangement he adopted, in his steam-carriage, in

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[1] See *A Descriptive History of the Steam Engine*, by Robert Stuart, 1824; pages 88, 90.

1828, is shown by fig. 325. A series of sections of 1¼-inch tubes arranged as a quadrilateral, with two transverse pipes at the front, are placed side by side, and form a furnace: the fire being laid on the lower pipes and the burning gases passing between the upper pipes to the chimney. The heated water circulates through the pipes in the direction of the upward inclination, and the steam is collected in a cylindrical reservoir above. The boiler was effective as a steam-generator.

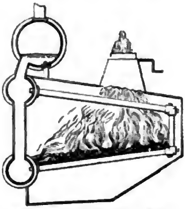
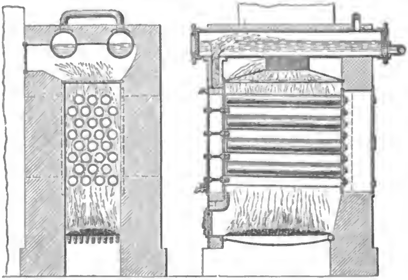


Fig. 325.—Sectional Steam Boiler, by Goldsworthy Gurney, 1825.

Dr. Ernst Alban, of Plau, Mecklenburg, was an earnest advocate of boilers on the tubulous principle. In 1843, he first published descriptions of his boilers and engines.[1] "The great principle, the sure method," he says, "is so to construct the boiler that its explosion may not be dangerous." The great principle is rather so to construct the boiler that it shall not explode at all. Dr. Alban's peculiar type of boiler, of 10 horse-power, is illustrated by figs. 326—a group of horizontal water-pipes in communication with a vertical water-space chamber, which opens into two reservoirs above; from which the steam is taken away. The boiler is filled with water up to the level of the centres of the reservoirs, the upper halves of which are filled with steam. The water-pipes, 28 in number, are of copper 4 inches in diameter, about $^1/_{10}$ inch thick, from 4¼ feet to 6¼ feet in length, according to requirements. They are closed at the back ends by a screw-cover, and are screwed to the back plate of the water-space chamber. Two openings through the plate into each pipe are made:—one below the centre of the pipe for the inflow of water; one above for the escape of steam into the chamber. The pipes are slightly inclined,—at the rate of ⅛ inch or



Figs. 326.—Sectional Steam Boiler, by Ernst Alban, 1843.

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[1] See *The High-pressure Steam Engine.* By Dr. Ernst Alban; translated from the German, with notes, by Dr. Wm. Pole, F.R.S., C.E.; 1848.

3/16 inch in one foot,—upwards towards the water-chamber, to facilitate the escape of steam from them. The pipes are arranged in eight rows, zigzag, so as to meet and divide the upward current. They are placed 1½ inches apart; but this distance may be increased with advantage, to facilitate the operations of cleaning. The water-space chamber is from 6 inches to 8 inches deep, of iron plates, ½ inch thick in front and ¾ inch at the back, strengthened with stay-bolts. Guiding-plates are fitted horizontally between the rows of pipes, to collect and divert the steam to one side of the chamber, and so keep the lower openings into the pipes clear for the water. The steam thus rises at one side of the chamber into the left-hand reservoir, whilst water descends from the right-hand reservoir into the chamber. The reservoirs are 12 inches in diameter, and 7 feet long. The fire-grate is 2 feet wide, and 4 feet long. The fire-bars are of cast iron, 1½ inches wide, with ½-inch air-spaces; concave at the top, in order to collect ash and protect the bars from excessive heat. The level of the grate is from 16 inches to 18 inches below the boiler, for coal as fuel; for wood, it is 6 inches lower; for turf or peat, 12 inches deeper. Dr. Alban's proportions give a consumption of coal at the rate of from 7 pounds to 10 pounds per square foot of fire-grate per hour. The area of the fire-grate is 8 square feet, or 4/5 square foot per horse-power. The heating surface of the pipes, taken as 4¼ feet long, is 125 square feet, and that of the lower halves of the reservoirs is 11 square feet; together 136 square feet, or 17 times the grate-area, or 13.6 square feet per horse-power.

THE BARROW SECTIONAL STEAM BOILER.

The Barrow sectional boiler, formerly known as the Howard boiler, was brought out in the year 1866, by Messrs. J. & F. Howard of Bedford. It was described by Mr. David Joy,[1] in 1875, as then constructed by the Barrow Ship-building Company; and it is illustrated in figs. 327, as now constructed by this company, showing a boiler of about 50 nominal horse-power. It consists of six vertical groups of six wrought-iron pipes in each, 9 inches in diameter externally, and 10 feet in length, ranged side by side, inclined upwards towards the back, at an angle of 1 in 8. The pipes of each group open into a vertical collecting pipe, 10 inches in diameter, at the back. The feed-water is introduced into a horizontal cast-iron chamber, square in section, laid transversely at the front, to which each of the six lowermost pipes is connected, and from which the feed-water is supplied to each group. The steam and water circulate in each pipe, and are separated in the vertical pipes at the back, from which the steam passes to the front and returns to the back, in the two uppermost rows of pipes which are above the water-line; and thence proceeds to the steam collector, which is a transverse cylinder of cast iron, 12 inches in diameter at the top of the furnace.

The wrought-iron water-pipes, figs. 328, are ¼ inch thick, with solid welded ends, ¾ inch thick. They are connected to the back collecting pipes by screwed neckings, and are closed at the front by cast-iron neckings screwed into the ends, closed with cast-iron plates fastened with four bolts and nuts, as shown. The transverse feed-water chamber is 4½ inches square, and is fixed in place by cast-iron plates, one opposite the end of each lowermost pipe, and four bolts and nuts. By removing these plates, direct access is obtained to the pipes. A steam reservoir is supplied, when required, erected immediately above the collector. It is 2½ feet in diameter and 9½ feet long.

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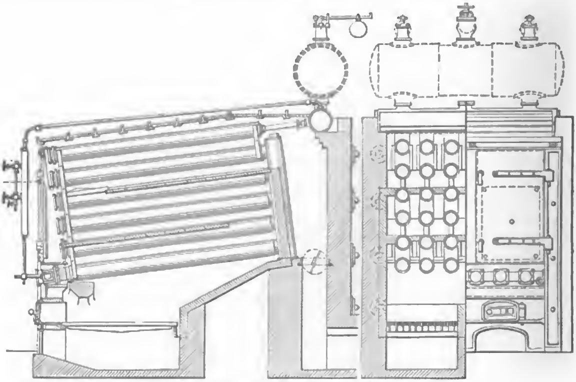
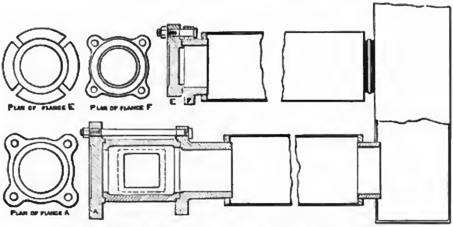
[1] See *The Journal of the Iron and Steel Institute*, 1875; pages 220, 387.



Figs. 327.—The Barrow Sectional Boiler. Scale 1/74th.



Figs. 328.—The Barrow Boiler: Pipes and Fittings. Scale 1/16th.

The boiler is inclosed in brickwork at the sides and the back, and by cast-iron covering-plates at the top and the front. The burning gases from the fire-place are conducted to the back of the boiler, thence to the front, and again to the back at the upper part, whence it descends to the chimney-flue underground. The brickwork externally is 15 feet long, 10 feet wide, and 13 feet high.

The evaporative power of the boiler is rated at one cubic foot of water evaporated from and at 212° F., per hour, for which 7 feet in length of pipe, or 16⅓ square feet of surface is allowed. The boiler shown in figs. 327 has 840 square feet of pipe heating surface, of which over one-third is steam-heating surface, exclusive of the surface of the back collecting chambers, which represents, say, 50 nominal horse-power, and an evaporation from and at 212° F., of 50 cubic feet of water per hour. The fire-grates, of which there are two, are each 5¾ feet long and 3½ feet wide, making together 40 square feet of grate-area, or $^1/_{21}$ part of the heating surface.

The weight of the boiler complete is 13¼ tons, or 5.3 cwts. per horse-power. The price, exclusive of a steam-receiver, is £410, or £8, 4*s*. per horse-power, or £31 per ton weight.

The boilers are tested to three times the working pressure, which is from 140 lbs. to 170 lbs. per square inch. The bursting pressure of the pipes is said to be over 1500 lbs. per square inch.

According to the statements of Mr. Joy, in the paper already referred to, it appears that in a boiler of 60 nominal horse-power, having, presumably, 1000 square feet of heating surface, and 48 square feet of grate-area, 698 pounds of Wigan coal was consumed, and 100½ cubic feet of water was evaporated per hour, in the ordinary course of work,—in the ratio of 9 pounds of water per pound of coal. Per square foot of fire-grate, the consumption of coal is 14½ pounds per hour, and of water 2.10 cubic feet per hour.

THE BABCOCK & WILCOX SECTIONAL STEAM BOILER.

This boiler is constructed by the Babcock & Wilcox Company, New York and Glasgow. Mr. Babcock states that the first person to use inclined water-tubes, connecting water-spaces, front and rear with a steam-space above, was Mr. Stephen Wilcox, his partner, in 1856.[1] Mr. Babcock succinctly states the advantages claimed for water-tube boilers, compared with shell boilers: 1. They admit of the use of very thin metal with ample strength; and such thin material may be exposed to the hottest gases comparatively with impunity. 2. A better circulation of water can be obtained in the water-tube boiler than in any other—a continuous rapid flow in one circuit. It keeps all parts at an even temperature, delays deposits, and sweeps away the steam as it is formed. 3. A water-tube boiler can be made sectional. 4. The life of a water-tube boiler may be indefinitely prolonged by simply replacing worn-out or burnt-out parts. The first boiler of the present form was constructed in 1869; it is still in use, and is good. 5. Ease of transportation. 6. Greater efficiency of heating surface, as the hot gases are caused to cross the pipes and impinge on the surfaces—an action which is more effective than the gliding of gases parallel to the surfaces.

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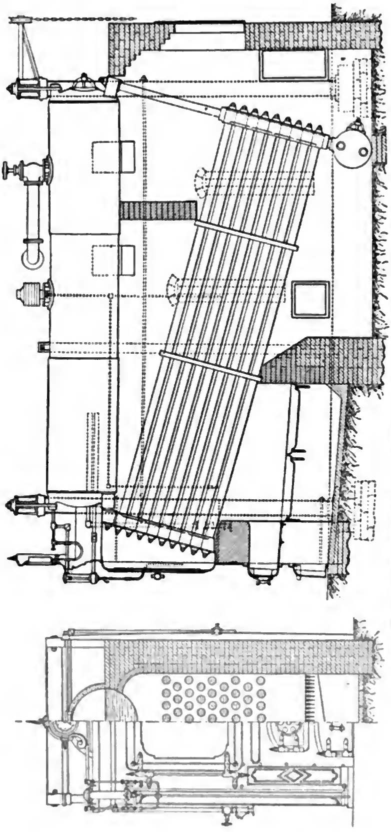
[1] See discussion of a paper on "Shell and Water-tube Boilers," by Mr. Allan Stirling, in the *Transactions of the American Society of Mechanical Engineers*, vol. vi. 1885; page 602.



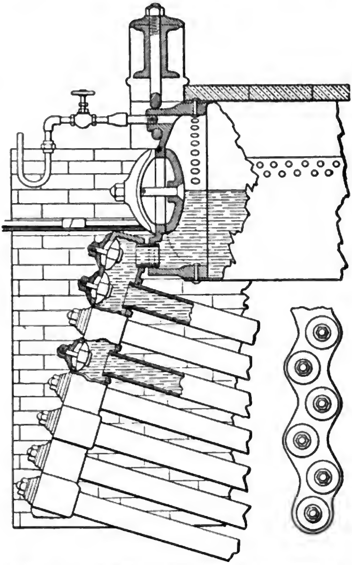
Figs. 329.—Sectional Boiler, of 120 Horse-power, by Babcock & Wilcox. Scale 1/64th.

7. The water-tube boiler is capable of being perfectly cleaned.

Mr. Babcock asks, "Why are not water-tube boilers in more general use? Because they require a high class of engineering to make them successful. The plain cylinder is an easy thing to make. It requires little skill to rivet sheets into a cylinder, build a fire under it, and call it a boiler; and because it is easy and any one can make such a boiler,—because it requires no special engineering—they have been made, and are still made, to a very large extent. The water-tube boiler, on the other hand, requires much more skill in order to make it successful."

The tubes or pipes of which the Babcock & Wilcox boilers are constructed are straight, of wrought iron, 4 ins. in diameter externally, and No. 9 wire-gauge, or $^5/_{32}$ inch in thickness, lap-welded. The length of the tubes varies according to the size of the boiler:—For boilers of from 10 to 45 horse-power actual, they are made 6, 8, 10, and 12 feet in length. For boilers of 45 horse-power and upwards, they are 14, 16, and 18 feet in length. The unit of horse-power, which is taken as the actual power for the production of which steam can be supplied, is the evaporation of 30 pounds, or nearly half a cubic foot, of water per hour. For this production, 11½ square feet of heating surface is provided. Heating surface is reckoned on the whole surface of the tubes, the ends of the boxes or headers, and half the surface of the steam drums. The quantity of coal consumed, when the boilers are working to their rated power, is from 10 pounds to 15 pounds per square foot of fire-grate. But occasionally as much as 30 pounds have been consumed.

A steam boiler of 120 horse-power is illustrated by figs. 329. The pipes are 18 feet in length, arranged in vertical sections of nine pipes to each section, making together 63 pipes. As the pipes of each section are zigzag, they in reality arrange themselves in 18 vertical rows. The zigzag arrangement facilitates the breaking up and mingling of the burning gases and the absorption of heat. The upper ends of the pipes are 4½ feet higher than the lower ends, or inclined at the rate of 1 in 4. The pipes are 12 inches apart vertically between centres, and the vertical rows are about 7 inches apart horizontally and diagonally. The clear spaces between the tubes are therefore equal to (7 − 4 =) 3 inches; and the outer tubes are 2 inches clear of the walls. The total width of the furnace fire-grate, and flue-chamber, between the walls is 53 inches; and the length is 20 feet 8 inches. The height of the flue-chamber to the bottom of the drum is about 11 feet 2 inches. The side walls are

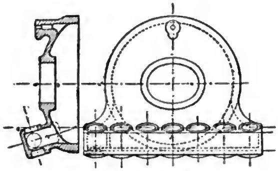


Figs. 330.—Babcock & Wilcox Boiler: Tube Connections and zigzag arrangement. Scale 1/26th.

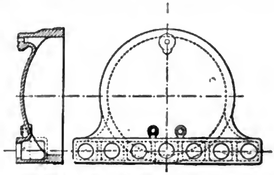
18 inches thick; the front wall is 20 inches thick, and the back wall 14 inches.

The tubes of each section are, as stated, zigzag, or "staggered," and they are fixed into cast-iron connections, boxes, or headers, at each end, as illustrated by figs. 330, by means of a roller tube-expander, a cored hole 4⅜ inches in diameter being formed in the casting opposite the end of each pipe, to admit the expander to the tube-end. These cored holes are closed each by a cover, jointed to the casting, metal to metal, held together by a wrought-iron clamp and bolt.

A horizontal drum of ⅜-inch plate, 3 feet in diameter, 18¼ feet long, is placed longitudinally and built in above the boiler. The longitudinal seams are double-riveted. It is connected to the headers of each section of tubes at each end: the connection at the lower or back ends of the tubes being made by a pipe about 4½ feet long, fixed by the expander. Thus a thorough continuous circulation of water and steam is maintained in the system. The boiler and drum are filled with water to the level of the centre of the drum, and so a large area of water surface, 18¼ feet by 3 feet, is provided. The mixture of steam and water rises from the upper end of the boiler into the drum, where the steam separates itself from the water, occupying the upper half of the drum as steam-space, and freeing itself from entangled water whilst it travels towards the other end of the drum to the steam-pipe, 5 inches in diameter. The water descends thence through the connecting tubes to the lower end of the boiler, whence it rises again towards the front end of the boiler. The heads of the steam-drum are of cast iron, 1 inch thick, except at the junction with the drum, where they are 2 inches thick. The front head, shown in figs. 332, is provided with connections and fittings for the steam-pressure gauge, and the glass water-gauge; the back head, figs. 331, is made with a manhole and cover. The sockets for the ends of the connecting pipes are shown. The safety-valve, 5 inches in diameter, and the stop-valve, also 5 inches, are fixed on the top of the drum.



Figs. 331.—Back Head.



Figs. 332.—Front Head.

Babcock & Wilcox Sectional Boiler: Drum-heads. Scale 1/32d.

The mud-drum at the lowermost part of the boiler is of cast iron, as the best material to withstand corrosion. It is 18 inches in diameter, 1 inch thick, and is provided with means for cleaning out. It is out of the course of the circulation, and is a receptacle for the mud and loose scale that fall from the boiler, with each section of which it is connected. The feed-water is introduced into the mud-drum.

The steam-drum and the boiler are suspended from two pairs of cross-girders, one at each end of the boiler, supported on columns, by links which pass between the girders of each pair, and take on two studs, one on

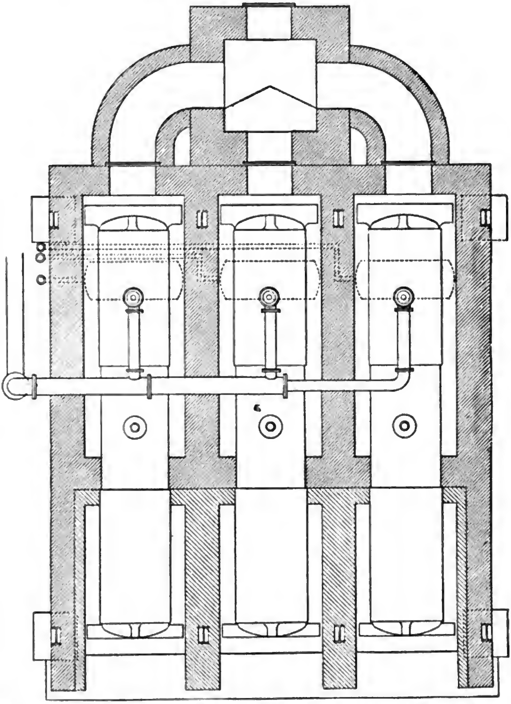


Fig. 333.—Babcock & Wilcox Boilers: Installation of three boilers of 120 horse-power each. Scale 1/64th.

each drum-head. By the adoption of this means of support, the connections are kept free from strains which might be caused by irregular bearings or by expansion; and the brickwork may be repaired or renewed, when necessary, without disturbing the boiler.

The pipes are tested and made tight under a hydrostatic pressure of 300 lbs. per square inch, iron to iron, without packing. The steam-drums are tested at a pressure of 150 lbs. per square inch.

The fire-grate is 53 inches wide, and 6 feet 1 inch wide, in two lengths of fire-bars, on a slope of 7 inches at the bridge below the level of the dead-plate. They make an area of 26.87 square feet of grate. To direct the current of hot gases effectively across the pipes of the boiler, the pipes are divided into three lengths by means of two transverse flame-bridges made of fire-brick, 2 inches thick, inserted in pieces between the pipes. The first of these flame-bridges rises from the ordinary bridge at the back of the grate; the second descends from an inverted bridge built down from the steam-drum. The flue at the back, to the chimney, takes its departure at a high elevation. By this means the hot current is caused to traverse the faggot of pipes three times before it passes to the flue. The headers at each end of the pipes fit closely together, and so constitute a partition which is substantially solid. The heating surface amounts to 1378 square feet, or 11½ square feet per horse-power, or 51.3 times the grate-area.

The brickwork is 23½ feet long, 7 feet 5 inches wide, and 11 feet high externally. The plan of the three boilers of 120 horse-power each, which were erected at the International Inventions Exhibition of 1884, for the generation of electric light, is shown in fig. 333, with the steam-pipe connections. A 5-inch pipe is taken from one of the outer boilers, and joins to a 7-inch main pipe which receives the steam from all the boilers.

The ordinary working pressure is 125 lbs. per square inch.

The shipping weight of each boiler is 3000 lbs., or 13.4 tons. The price is £403, or £3, 9*s*. per horse-power, or £30, 15*s*. per ton of shipping weight.

The boilers of the Brooklyn Sugar Refining Company are shown in general elevation, in fig. 334, with the fuel economizer between the boilers and the chimney.

The 10-horse-power Babcock & Wilcox boiler is illustrated by figs. 335. It is composed of three sections of tubes, 6 feet long, four tubes in each section; in all, 12 tubes. The drum is 2 feet in diameter, of ¼-inch plate, 6 feet in length. The fire-grate is 2 feet 1 inch wide, and 2½ feet long, making an area of 5.21 square feet. The heating surface is 118 square feet, or 22.6 times the grate-area.

Table No. 158 gives the general data for a selection of boilers of various powers, from 10 to 480 horse-power.

The Babcock boilers at the Inventions Exhibition, in the course of regular duty, evaporated from 11 pounds to 11½ pounds of water per pound of coal; the feed-water being supplied at 190° F. The results of many tests recorded in *Steam*, a pamphlet issued by the constructors, show that, as averages of 20 different tests, with all classes of fuel, 15.4 pounds of fuel was consumed per square foot of fire-grate, 11.29 pounds of water was evaporated per pound of combustible, as from and at 212° F., and 3.71 pounds per square foot of heating surface.

Table No. 158.—BABCOCK & WILCOX WATER-TUBE STEAM BOILERS.

Horse-power.	Tubes.			Drums.					Shipping Weight.	Space Occupied.						Safety-valves.		Steam-pipe, Diameter.	Furnace.					Heating Surface.	Ratio of Heating Surface to Grate-area.
	Number of Sections.	Tubes in each Section.	Length.	Number.	Diameter.	Thickness.	Length.			Length.		Width.		Height.		Number.	Diameter.		Width.		Length.		Area of Fire-grate.		
In One Furnace.																									
H.P.	secs.	tubes.	feet.	drums.	inches.	inch.	ft.	ins.	lbs.	ft.	ins.	ft.	ins.	ft.	ins.	valves.	inches.	inches.	ft.	ins.	ft.	ins.	sq. ft.	sq. ft.	ratio.
10	3	4	6	1	24	¼	6	0	6,700	9	3	4	5	7	7	1	2	2½	2	1	2	6	5.21	118	22.6
15	3	5	8	1	24	¼	8	0	8,200	11	3	4	5	8	1	1	2	2½	2	1	3	6	7.29	181	24.8
25	4	5	10	1	30	5/16	10	0	11,000	13	3	5	0	8	7	1	2½	3	2	8	3	6	15.35	287	18.6
35	4	6	12	1	30	5/16	12	0	13,600	15	3	5	0	9	1	1	3	3	2	8	4	6	12.02	402	33.1
45	4	7	14	1	30	5/16	14	4	16,500	19	0	5	8	12	3	1	3½	3½	2	8	6	0	16.00	512	32.0
51	4	8	14	1	30	5/16	14	4	17,000	19	0	5	8	12	9	1	3½	3½	2	8	6	0	16.00	585	36.5
73	6	7	16	1	30	5/16	14	4	22,000	21	0	6	10	12	3	1	3½	3½	3	10	6	0	23.00	840	36.5
104	6	9	18	1	36	⅜	18	8	28,000	23	0	6	10	13	9	1	5	5	3	10	7	0	26.83	1193	44.5
120	7	9	18	1	36	⅜	18	8	30,000	23	0	7	5	13	9	1	5	5	4	5	7	0	30.91	1378	44.6
184	12	9	16	2	30	5/16	16	6	46,000	21	0	10	4	13	3	2	4	6	7	4	7	0	51.33	2111	41.1
240	14	9	18	2	36	⅜	18	8	55,000	23	0	11	6	14	0	2	5	8	8	6	7	0	59.50	2756	46.3
In Two Furnaces.																			Each Furnace.						
90	8	7	14	2	30	5/16	14	4	30,000	19	0	9	10	12	3	2	3½	3½	2	8	6	0	16.00	1025	32.0
102	8	8	14	2	30	5/16	14	4	32,000	19	0	9	10	12	9	2	3½	3½	2	8	6	0	16.00	1171	36.5
122	10	8	14	2	30	5/16	14	4	35,000	19	0	11	0	12	9	2	3½	3½	3	3	6	0	19.50	1411	36.0
184	12	9	16	2	30	5/16	16	6	45,000	21	0	12	2	13	3	2	4	4	3	10	7	0	26.83	2111	39.3
240	14	9	18	2	36	⅜	18	8	57,000	23	0	13	4	14	0	2	5	5	4	5	7	0	30.91	2756	44.6
480	28	9	18	4	36	⅜	18	8	108,000	23	0	22	0	14	0	2	5	8	8	6	7	0	59.50	5513	46.3

NOTE.—The lengths of fire-grate given in this table are those adopted for the United States. For English practice they are less.

A boiler of 136 horse-power at Grant's Mills, Ramsbottom, was tested for evaporation, in July, 1884. The fire-grate was 5 feet wide, 6 feet long, having 30 square feet of area. The heating surface was 1563 square feet, or 52 times the grate-area. The first test was made with Scholes & Co.'s nut coal at 8*s.* per ton, the second and third with dross, Pickup at 4*s.* 9*d.* per

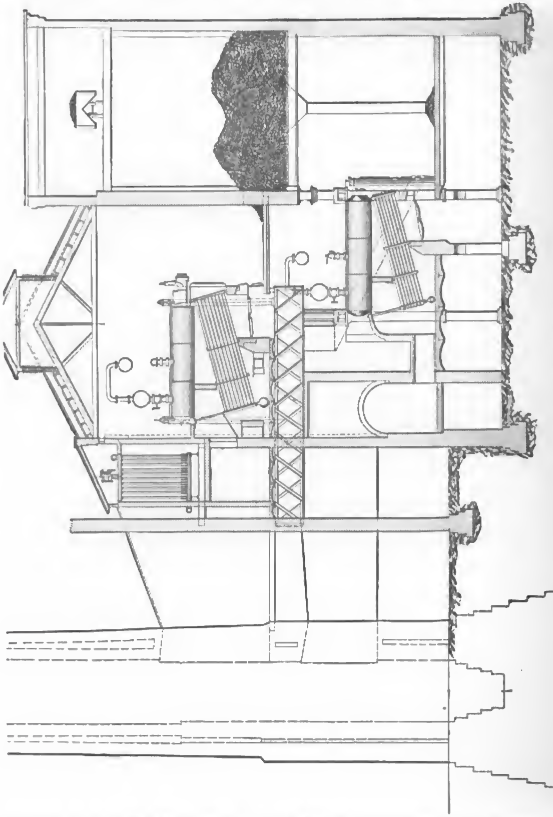
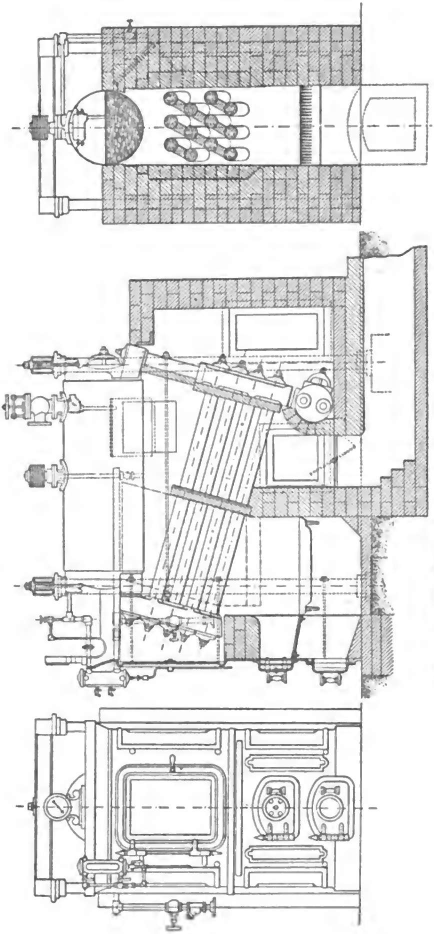


Fig. 334.—Boiler-house of Brooklyn Sugar-refining Co., with Babcock & Wilcox Boilers, Brooklyn, N.Y. 1000 H.P. erected 1877; 500 H.P. erected 1879; 500 H.P. erected 1882; 1464 H.P. erected 1884.



Figs. 335.—Babcock & Wilcox Sectional Steam Boiler, of 10 horse-power. Scale 1/38th.

ton and Crosser at 5*s.* 3*d.*, in equal parts mixed. Each test lasted eight hours.

Table No. 159.—BABCOCK & WILCOX BOILER, RAMSBOTTOM: RESULTS OF TRIALS, 1884.

No. of test	1	2	3
Average pressure of steam	85 lbs.	65 lbs.	90 lbs.
Temperature of feed-water by injector	210° F.	208°	208°
Coal consumed per hour	658 lbs.	728 lbs.	714 lbs.
Do. do. per square foot of grate	21.90 „	24.26 „	23.80 „
Ash per cent	8.4%	11%	11.7%
Water evaporated per hour	6125 lbs. 102.1 cu. ft.	6912 lbs. 115.2 cu. ft.	7612 lbs. 123.9 cu. ft.
Do. do. per sq. foot of grate	204 lbs. 3.40 cu. ft.	230.4 lbs. 3.84 cu. ft.	280.6 lbs. 4.67 cu. ft.
Do. per pound of coal, actual.	9.14 lbs.	9.50 lbs.	10.66 lbs.
Do. do. do. from and at 212° F.	9.40 „	9.84 „	11.17 „
Do. per pound of combustible, do.	10.38 „	11.03 „	12.65 „
Horse-power developed, at the rate of 30 lbs. of water evaporated from 212°, at 70 lbs. pressure per hour	205.8 H.P.	232.2 H.P.	258 H.P.

A steam boiler of 140 horse-power, at the works of Messrs. J. & J. M. Worrall, Manchester, fitted with a special 4-feet drum, and Jucke's stoker, was tested in July and August, 1885. The fire-grate was 6½ feet long, 4½ feet wide, making 29.25 square feet of area. The heating surface was 1616 square feet, or 55 times the grate-area. The leading results are given in the following table:—

Table No. 160.—BABCOCK & WILCOX BOILER, MANCHESTER: RESULTS OF TRIALS, 1885.

No. of test	1	2
Duration of test	27 hours	18½ hours
Average pressure of steam per square inch	75 lbs.	95 lbs.
Temperature of feed-water by injector	135°	275°
Coal consumed per hour	946 lbs.	633 lbs.
Do. do. per square foot of grate	32.3 „	21.6 „
Water evaporated per hour	8007 „ 130.2 cu. ft.	6400 „ 110.3 cu. ft.
Do. do. per square foot of grate	273 lbs. 4.44 cu. ft.	219 lbs. 3.77 cu. ft.
Do. do. per pound of coal, actual	8.46 lbs.	10.125 lbs.
Do. do. do. from and at 212° F.	9.42 „	9.85 „
Horse-power developed, at the rate of 30 lbs. of water evaporated from 212° at 70 lbs. pressure per hour	297 H.P.	208 H.P.

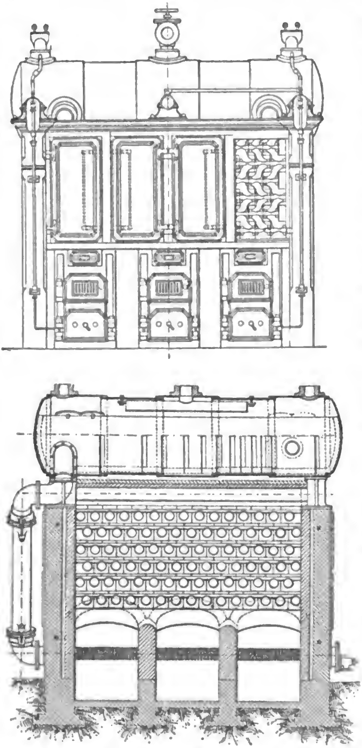
THE ROOT SECTIONAL STEAM BOILER.

The Root boiler, it appears, was introduced in America, a year later than Babcock & Wilcox's boiler. The leading features of the Root boiler have already been noticed, pages 256 and 268. But, recently, in its latest form, it has been materially improved. The boiler, formerly partly filled with water and partly occupied with steam, is now filled up with water, which rises into and occupies one half of the transverse reservoir or drum on the top of the boiler,—the upper half of the receiver being occupied by steam. On this system, the steam and water rise into the receiver, where they separate, and whence the water descends direct to the transverse mud-pipe at the back. Thus, a continuous circulation of water and steam is established.

The unit of the Root boiler is a wrought-iron pipe 4½ inches in diameter externally, No. 8 wire-gauge or nearly 3/16 inch thick, and having 10 feet 4 inches of clear length of heating surface between the cast-iron heads into which it is screwed.

A boiler of 100 nominal horse-power is illustrated by figs. 336 and 337. There are fifteen vertical sections of tubes, placed zigzag, six tubes in each section, making 90 tubes altogether. The ends of the tubes of each section are screwed each into a rectangular box-casting at each end, and these castings are connected by bracket-like hollow castings, as shown in figs. 338, which are means of communication and circulation from tube to tube. The rectangular castings abut on each other, and they form a compact wall or partition, by which the furnace-chamber is inclosed. Thus the bracket-junctions and their fastenings are protected from the direct furnace-heat. The tubes of each section are 10 inches apart obliquely, between centres; and horizontally the tubes are 8 inches apart: forming a mass of tubes 4 feet in total depth. They are inclined upwards towards the front at a slope of 1 in 4, being 28 inches higher at the fore end than at the back. The current of water and steam rises to the front, and is delivered from each section into an upper collecting pipe of cast iron, 8 inches in diameter, from which it passes by two 8-inch delivery-pipes into the drum at each end. The drum is of ½-inch plate-iron, 3½ feet in diameter, and 13½ feet long. It is double-riveted in the longitudinal seams, with 5-inch lap and 2¼ inches pitch; single-riveted transversely, with 2-inch lap and 2-inch pitch. The water passes from the drum at the other end, downwards, through an 8-inch pipe, into the mud-drum, a transverse 9-inch cast-iron pipe, which is placed horizontally at the back of the boiler, at the lowest level. From this pipe, the water rises by branch pipes into each section of tubes, and so continuous circulation is maintained. A 6-inch perforated pipe for collecting the steam from the drum is fastened to the upper part; it delivers the steam direct to the steam-pipe and stop-valve. The working pressure is 150 lbs. per square inch. The boilers are tested to a pressure of 600 lbs.

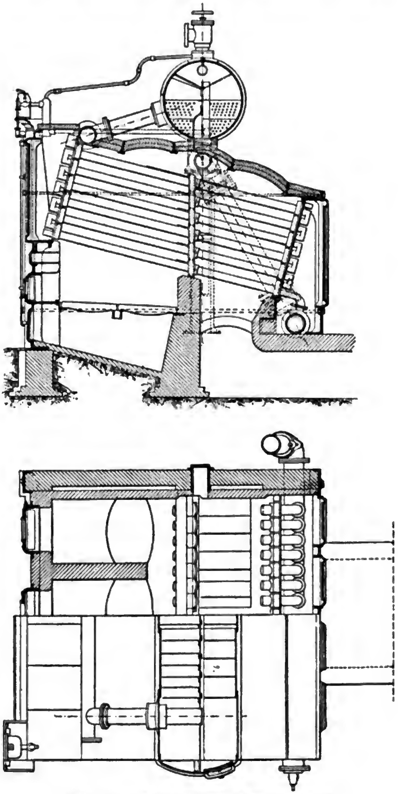
The boiler is inclosed by 14-inch brick walls, 10 feet apart; and at the front and the back by cast-iron plates. It is roofed over by arched fire-slabs in two thicknesses, making together 6 inches, supported on cross girders. The roof slopes in order to economize brickwork, and also to minimize the upper flue-space. There are three fire-places with grates, each 2 feet 10 inches wide, and 5 feet 3½ inches long, in two lengths of fire-bars, making together an area of 45 square feet of grate. The fire-bars are ½ inch, with ⅜-inch air-spaces. An opening can be made above the doorway for the admission of air. The ash-pit is closed by an ordinary ash-pit door. One flame-bridge or partition is erected over the ordinary furnace-bridge, so as to direct the draught directly upwards from the grate, traversing the water-tubes, then downwards again across the water-tubes, into an underground flue leading to the chimney. The flame-bridge consists of cast-iron blocks, one on each tube, held in place by a clip bolted to the tube.



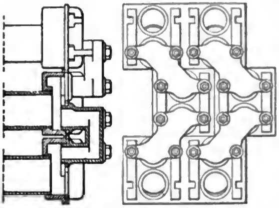
Figs. 336.—Root Sectional Steam Boiler. Scale 1/64th.

The heating surface of one tube is 12 square feet, which is taken as the unit of nominal horse-power. For 90 tubes the surface is 1080 square feet. Adding 65 square feet for the exposed area of the heads at both ends of the pipes, a total heating surface of 1145 square feet is obtained. Strictly, this area, for the given nominal power, 100 horses, allows only $\left(\frac{1145}{100}=\right)$ 11.45 square feet per horse-power; and it is equal to 25.5 times the grate-area. Evaporation at the rate of 50 pounds of water per nominal horse-power per hour, with 12 square feet of heating surface, can be effected.

The steam-drum is fitted as Stollwerck's water purifier. The lower half of the drum, occupied by water, is divided longitudinally into two compartments, each of which is subdivided transversely by a number of perforated diaphragms. The feed-water is introduced into the drum at one end of one of the compartments at the back of the circulating downflow pipe, where it meets the hot water and steam delivered into the drum at that end: mixing with the hot water, and slowly travelling through the perforated diaphragms to the other end. Thence it crosses over to the other compartment, where it is met by the inflowing hot water and steam, delivered at the other end by a cross-over pipe shown in the plan, figs. 337. Travelling slowly back by the second compartment to the first end of the drum, the feed-water, thoroughly mixed with the other water of the boiler, passes away by the downflow pipe, in the regular course of circulation.



Figs. 337.—Root Sectional Steam Boiler. Scale 1/64th.



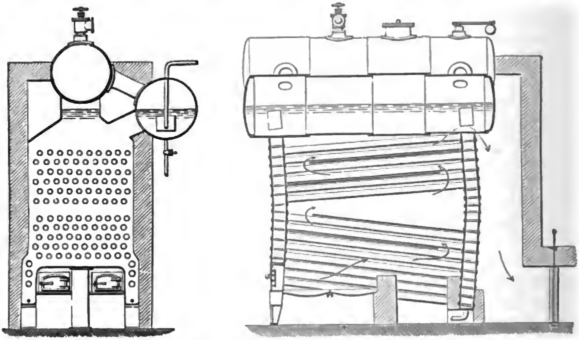
Figs. 338.—Root Boiler: Junctions of Tubes. Scale 1/16th.

In the course of its passage along the drum, the feed-water, heated approximately to the temperature of the steam, drops the salts of lime and magnesia, which are insoluble in hot water, between the perforated diaphragms. The deposits are cleared out from time to time through manholes provided for the purpose.

The nominal horse-power of the "new model" Root boiler is measured by 10⅔ square feet of heating surface. The weight of the boiler is 22½ tons, and the price is £920, or about £41 per ton of weight.

THE SINCLAIR SECTIONAL STEAM BOILER.

In the Sinclair boiler, figs. 339, the pipes are straight, and are arranged in two groups of several rows in each. They open into two flat stayed water-space chambers, one at each end, nearly vertical, and forming a front and a back to the boiler. The sides, top, and back are inclosed by brickwork. The water-space chambers open into a horizontal cylinder at the upper part, at one side of the structure, which is above the level of the pipes, and is kept half full of water, as a storage cylinder, so that all the pipes are full. A second cylinder is placed directly above the pipes, and is used as a storage reservoir for steam. The fire-grate is placed underneath the system of pipes; and there are three other pipes at each side of the



Figs. 339.—Sinclair Sectional Steam Boiler.

furnace, to guard the brickwork. Access is had to the pipes, through a round hole cut in the outer plate of each flat water-chamber, opposite the ends of each pipe, as in fig. 340. Each hole is fitted with a conical cover on the inside of the plate, drawn up to it by a central bolt passing through a cast-iron bonnet on the outside. The cover takes its bearing on the inner edge of the hole, making an effective joint.

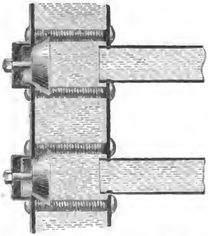


Fig. 340.—Sinclair Boiler: Pipe Fittings.

The pipes are slightly inclined, the lower group inclined upwards towards the front, over the fire, and the upper group inclined upwards towards the back. The water and steam ascend in each group into the water-spaces, whence they pass into the first cylinder, where the steam is separated and whence it passes into the reservoir at the top.

The pipes are divided by deflecting plates or partitions, so as to deflect the burning gases rising from the fire-grate, towards the back and front alternately of each group—making five traverses over and between the pipes,—after which the gases pass off into a descending flue at the back of the boiler; thence to the chimney.

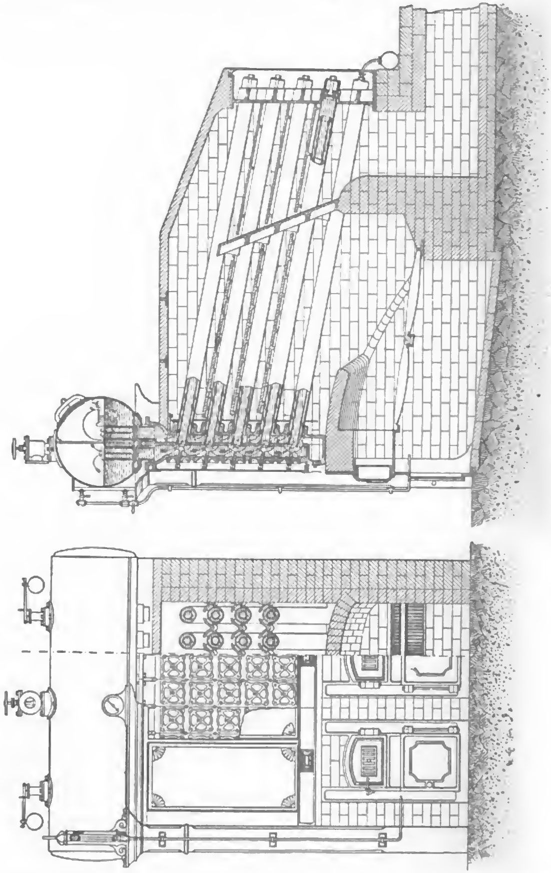
The Sinclair boiler of 75 nominal horse-power, illustrated by figs. 339, has 115 water-pipes, lap-welded, 11¾ feet in length. Of these pipes, 109, which form the two groups above the fire, are 4 inches in diameter, and ⅛ inch thick; and the six others, flanking the furnace, are 5 inches in diameter. The two storage cylinders, for water and steam, are formed with dished ends; they are 4 feet in diameter and 16¾ feet long. The fire-grate is 6 feet 7 inches wide and 6 feet in length. The length outside, over the brickwork, is 18 feet, the width 9½ feet, and the height 18 feet.

The area of fire-grate is 39.5 square feet, and that of the heating surface is 1507 square feet, or 38.1 times the grate-area. These quantities are at the rate of about ½ square foot of grate-area and 20 square feet of heating surface, per nominal horse-power; evaporating 1 cubic foot of water per hour. The weight of the boiler complete is about 24¼ tons; or, without boiler mountings, 20½ tons. The weight complete is at the rate of about 6½ cwts. per horse-power. The price of the boiler complete is £700, equivalent to £9, 6*s*. 8*d*. per horse-power, or to £29 per ton of weight.

THE LANE SECTIONAL STEAM BOILER.

The Lane boiler, illustrated by figs. 341, is distinguished by two features: —one, the furnace, which is constructed with a deflecting arch of fire-brick, by the aid of which combustion is promoted and smoke prevented in some degree; and the circulating tube inside the water-tubes, in connection with vertical diaphragm chambers and a receiver at the front, for inducing rapid circulation of the water.

The unit of the Lane boiler has been a water-pipe, 5 inches in diameter externally, ¼ inch thick, and 10 feet in length, of which a length of 9 feet,



Figs. 341.—The Lane Sectional Steam Boil Scale 1/40th.

between the cast-iron fixtures,—is exposed as heating surface. The tubes are inclined upwards towards the front, at an angle of about 18 inches in 9 feet, or 1 in 6. The internal circulating tube is 2¾ inches in diameter outside, No. 12 wire-gauge, or about $^1/_{10}$ inch in thickness. It is placed eccentrically within the water-pipe,—nearer the top than the bottom. The water-pipes of each vertical section are fixed at the front ends into the back of an upright square chamber, which is divided by a transverse vertical partition into two compartments. The pipes are closed at the lower ends by cast-iron plugs. The circulating tubes are fixed into the partition at the front ends, and open into the front compartment; and they reach down the water-pipes to a clear distance, at the lower end, of 6 inches from the bottom. With this combination, rapid circulation of water and steam, under a head of 8 feet, takes place,—water from the front compartment of the vertical chambers down the circulating tubes, and a mixture of hotter water and steam up the water-pipes around the circulating pipes into the back compartment of the chamber. A mud-box is formed at the base of each chamber. A transverse blow-off pipe is laid at the back of the boiler, connected to the lowest tier of pipes. The heated water and steam rise through the inner halves of the vertical chambers into a cylindrical reservoir placed transversely on the top of the boiler; and the descending currents flow down the outer halves of the chambers. The reservoir is 2 feet 9 inches in diameter, and 8 feet 7 inches long, outside, of $^7/_{16}$-inch plate, double-riveted longitudinally, with $^9/_{16}$-inch plate for the ends. The ends are dished with a versed sine of about 2 inches, and are flanged to join the body of the receiver. The reservoir is divided by a longitudinal partition, directly over the partitions which divide the upright water-chambers. The partition does not reach to the bottom of the receiver, so permitting the water to keep the same level on both sides of it. The normal water-level is that of the centre of the receiver. The front and back compartments of the water-chambers are prolonged by pipes which rise into the receiver, and are slotted to permit the circulation of the water through their sides. The delivery-pipe is furnished with a semicircular bend, to deflect and throw down the current of water discharged, whilst the intermixed steam is separated. In the getting up of steam, it is observed that the water simply circulates through the slots; but when steam is formed, the mixture of steam and water rushes into and is discharged from the bend.

There are, as a rule, five pipes in each vertical section, and as each pipe presents 11.8 square feet of heating surface, the group of five pipes together presents 59 square feet. The number of vertical sections provided is according to the boiler power required. In the present instance, there are eight sections of pipes, making 40 pipes in all; and the total heating surface is ($59\times8=$) 472 square feet. There are two furnaces, each 35 inches wide, and 6 feet long as measured to the bridge. But the grate is only 5 feet long, in two lengths of bars. The area of each grate is 14.6 square feet; for two grates, 29.2 square feet. The heating surface is 16.2 times the grate-area. The working pressure is, by preference, from 80 lbs. to 100 lbs. per square inch, but it may be as high as 125 lbs.

The unit of nominal horse-power is the evaporation of 60 pounds of water per hour, from and at 212° F., and for this duty 10 square feet of heating surface is usually allowed. The nominal power of the boiler is, therefore, (472 ÷ 10 =) 47.2 horse-power, although taken as 50 horse-power, and the evaporative power is at the rate of (47.2 × 60 =) 2832 pounds, or 47.2 cubic feet of water, per hour, from and at 212°. The grate-area is at the rate of .62 square foot per nominal horse-power. According to the results of rough tests, 8½ pounds of water has been evaporated per pound of coal; or 7 pounds per pound of common slack. Taking a mean rate of 8 pounds per pound of fuel, (2832 lbs. ÷ 8 =) 354 pounds of coal is consumed per hour, or 12.1 pounds per square foot of fire-grate per hour.

Mr. Lane states as the result of his experiments, that for sectional boilers of all kinds, a thin bright fire is the best,—making as much of radiant heat as possible. Each fire-grate, in the present instance, is overarched, for a length of 21 inches, with fire-bricks. Arches extending 3 feet over the fire-grate have been tried, and it was found that, though the combustion was better, with less smoke, than with the shorter arch, or in the absence of an arch, yet the evaporative efficiency of the fuel was less; and he, with reason, ascribes the difference to the forced inaction of the radiant heat of the fire, in the case of the long overarching. At the same time, Mr. Lane says, it is not advisable to dispense altogether with the arch, as smoke is produced, and cold air rushes amongst the tubes without taking part in combustion. "We can work," he adds, "entirely without smoke by making the furnace very deep at the back, and by having the arch 3 feet in extent. To work in this way, however, causes a loss of efficiency and economy in radiant heat not utilized." The deflector, or arch, he finds, "distributes the heat over the tubes more uniformly: any one part does not get too much, also the gases cannot rush up and down and away."

The group of water-tubes is divided into two parts by a flame-bridge, constructed of cast-iron blocks laid in, for the direction of the burnt gases; the flame-bridge rising from the ordinary furnace-bridge. The gases rise and pass upwards through the first part of the group, and then descend through the second part to the chimney-flue which is constructed on the ground. In order to deflect the ascending hot currents upon and over the water-tubes, deflector-tiles, shown in detail, fig. 342, are laid over and between the tubes; and similar deflectors of cast-iron are laid on the tubes at the back of the bridge.

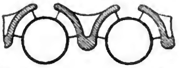


Fig. 342.—The Lane Boiler: Deflector-tiles over Pipes. Scale 1/16th.

The boiler is inclosed in a space 5 feet 10 inches in width, between two 14-inch brick walls. The top is roofed in with 3-inch fire-tiles, supported by cast-iron girders. The front and back are encased with cast-iron plates.

In his most recent practice, Mr. Lane makes the tubes 12 feet in length, instead of 10 feet; he has also placed four additional tubes at the sides of the furnaces, one at each side of each furnace, making 44 tubes in all. The weight of the boiler is about 10 tons, and the price is £375, or £37, 10*s.* per ton. This boiler is reckoned to be of fully 50 nominal horse-power.

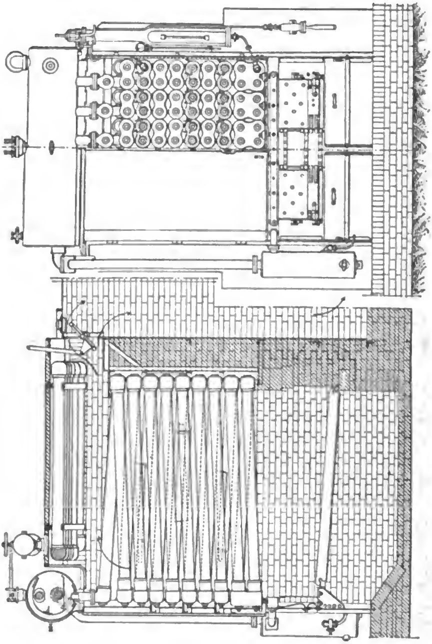
THE BELLEVILLE SECTIONAL STEAM BOILER.

The Belleville boiler, like the other sectional boilers already described, has a wrought-iron tube for its unit of construction. It is constructed in three series, according to the size of the tube:—82 millimetres, 100 millimetres, and 125 millimetres, internal diameter; or 3.23 inches, 3.94 inches, and 4.92 inches; and 7 millimetres, or .276 inch in thickness. The tubes are fitted together in vertical sections (*éléments amovibles*) placed side by side within a furnace-chamber, into each of which the feed-water is admitted at the lower end, and from the upper end of which a mixture of steam and water is discharged into a horizontal drum (*collecteur-epurateur*) above. The tubes of each section do not receive separate supplies of feed-water, as in the English and American sectional boilers. They are combined spirally, in such a manner that the feed-water is supplied only to the lowermost tube, and winds its way upwards through the tubes successively, in company with the steam that is generated from it in the course of its progress. The boiler is only filled about half-way up with water, the upper tubes being occupied by steam only, with such water in a state of minute division as may rise in mixture with it. In the drum, the water is separated by centrifugal action from the steam; and thence the steam passes into a serpentine group of drying pipes (*sécheur de vapeur*) in the uppermost part of the inclosure, to evaporate such moisture as may pass with it from the drum. Thence it passes into the main steam-pipe.

The drum is fitted as a purifier, in which lime-salts in the feed-water are precipitated on the principle already explained in connection with the Root boiler,—the precipitation of calcareous matter insoluble in hot water. This precipitate in a minutely subdivided condition is carried by the circulating water into a settling receiver (*récipient-déjecteur*), where it falls to the bottom, whilst the clear water passes into the supply pipe (*tube-collecteur d'alimentation*) to feed the boiler.

The unit of nominal horse-power is the evaporation of 20 kilogrammes of water, or 44 pounds of water, per hour.

An example of the Belleville boiler, of 100 nominal horse-power, for stationary use, is given in figs. 343 and 344. It contains 108 boiler tubes, 4.92 inches,—nearly 5 inches,—in diameter internally, .276 inch thick; making a diameter externally of 5½ inches. The tubes are disposed in six sections, of nine pairs of tubes, or 18 tubes each. They are screwed into sockets formed on cast-iron heads at each end; and are jammed by rings screwed up against the ends of the sockets, with metallic mastic. Each cast-iron head forms a connection between two pipes, and thus a continuous spiral thoroughfare from the bottom to the top of each section is established. The pipes are 6 feet 8 inches in clear length between the sockets. The inclination of the pipes is 3½ inches in their length, or 1 in 23. Two openings are made in the face of each head, opposite the end of each tube, through which the tubes may be inspected and cleared out when necessary. They are closed by doors fastened by anchor bolts and nuts, as shown in section in fig. 345. The steam that is formed in each section rises into, and enters at the bottom of, the drum, to which the rising pipe is connected with a simple conical joint. The drum is placed transversely above the boiler, out of reach of the gaseous products. It is shown in section in figs. 346. It is 19½ inches in diameter and 6 feet 10 inches long, of plates nearly ½ inch thick. The rising pipes are, in these views, shown as screwed into the bottom of the drum. The steam with the entrained water are delivered into an annular space formed within the drum by a circular partition, and pass round the interior of the drum; so developing a centrifugal action which determines the separation of the steam from the water and other foreign matter which may be intermixed.



Figs. 343.—The Belleville Sectional Steam Boiler. Scale 1/50th.

The quantity of water thus retained in the drum is from four to eight times as much as the feed-water, which also is discharged into the drum at the right-hand end. The feed-water is delivered in a divided state amongst the steam, and the rapid elevation of its temperature in contact with the steam causes instant precipitation of the calcareous salts in a pulverulent condition. The fresh feed-water with the precipitate in suspension, meets and mixes with the entrained water on the flat bottom of the partition, and they run off together at the left-hand end of the drum, down the connecting pipe to the settling receiver, which is 10 inches in diameter, and 39 inches deep. Here the solids are steadily deposited, to be drawn

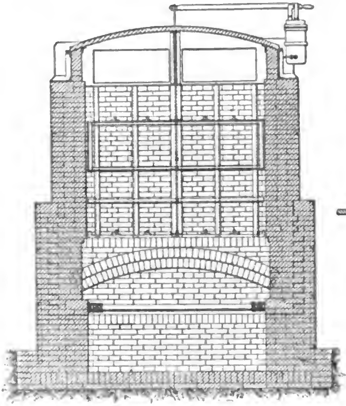


Fig. 344.—Belleville Sectional Boiler. Scale 1/50th.

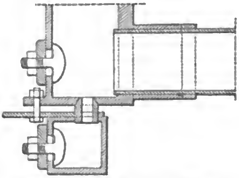
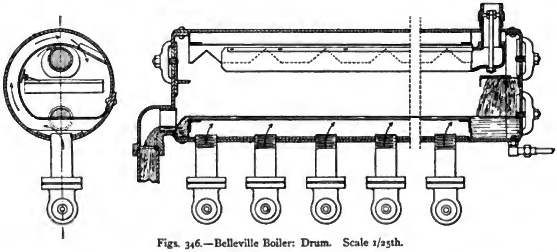


Fig. 345.—Belleville Boiler: Pipe Fittings. Scale 1/10th.



Figs. 346.—Belleville Boiler: Drum. Scale 1/25th.

off by a tap at intervals; and the clear water passes off by a short connection at the top of the receiver, into the supply-pipe, or conduit, 4½ inches square, from which it passes up into the boiler. The conduit is connected to the cast-iron heads with conical joints, which can be broken with facility, as shown in fig. 345, which is a vertical section of part of the lowest cast-iron head of one of the sections, with a part of the water-tube, and the feed-water conduit.

The steam-dryer, into which the steam passes from the drum, consists of twelve tubes, of the same diameter as the water-tubes, 6 feet in length, laid horizontally close to the roof of the fireplace, side by side, and connected by cast-iron heads at each end alternately in serpentine fashion. The steam takes a zigzag course through the pipe, and passes out to the stop-valve and steam-pipe.

The feed-water supply is regulated automatically by means of a float acting through an intermediate lever encased with the float, on an external weighted lever linked to the stem of a conical valve, detailed in figs. 347 and 348, by which the feed is regulated. The float is 5½ inches in diameter, and acts on the regulating valve with a leverage of 6½.

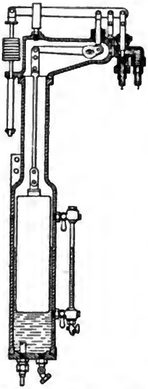


Fig. 347.—Belleville Boiler: Feed-water Regulator. Scale 1/20th.

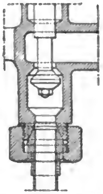


Fig. 348.—Belleville Boiler: Detail of Conical Valve of Feed-water Regulator. Scale ¼th.

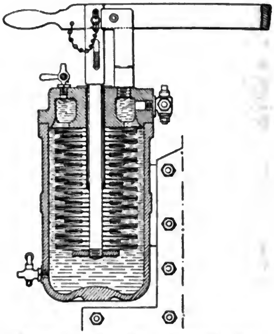
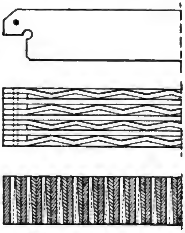


Fig. 349.—Belleville Boiler: Pressure Regulator. Scale 1/10th.

The rate of combustion and the steam-pressure are controlled automatically by means of an upright pressure cylinder, 8½ inches in diameter, which is filled with water, fig. 349, in communication with the boiler and subject to the pressure of the steam. A spindle, 1 inch in diameter, passes through the cover and forms a connection between a spring within the cylinder and a lever outside, the longer arm of which is pinned to a swivelling damper. The actuating pressure is measured by the sectional area of the spindle; and it acts through a reducing lever, in the ratio of about 30 to 1, on the lever fixed on the damper-shaft.

The boiler is inclosed in a furnace-chamber 6¼ feet wide, between two 14-inch brick walls, with a back wall of brick, 14 inches thick behind the boiler, and 20 inches at the back of the furnace. The fire-grate is about 7 feet long and 6¼ feet wide, making an area of 43¾ square feet. The products of combustion rise between the water-tubes, partially baffled by deflecting plates, till they reach the upper part of the chamber, enveloping the steam-drying pipes and passing over to and through the swing-dampers into the descending flue to the chimney. The fire-bars are formed alternately straight, ½ inch thick, and zigzag, 13/32 inch thick, as in figs. 350. The bars thus placed in frequent contact are solidly disposed, and support each other. The triangular interspaces for air thus formed are 13/32 inch wide at the apex. By this design, it is stated, the bars are kept comparatively cool, and the adhesion of clinker is prevented.

Steam is discharged in jets over the grate, in order to improve combustion and minimize smoke.



Figs. 350.—Belleville Boiler: Fire-bars. Scale 1/10th.

The Belleville boilers are stamped for a pressure of 12 atmospheres, or 165 lbs. working pressure per square inch. The working pressures vary from 105 lbs. to 165 lbs. Such high pressures are chosen in order, with a high temperature, to effect the separation of the salts in solution. The 100-horse-power boiler is guaranteed to evaporate 4620 pounds, or 74 cubic feet, of water per hour.

The jury of the Nice International Exhibition of 1883–84 state in their report results of official trials of the Belleville boilers on the advice-boat Voltigeur, using Anzin briquettes:—

	Water per Pound of Fuel.
Fires urged to their maximum................	7.31 lbs.
Fires urged for the normal power............	8.30 „
Fires working slowly............................	8.92 „

The jury also give results of comparative tests, made at Paris, of a Belleville boiler using semi-bituminous Anzin coal at 14*s.* 6*d.* per ton, and a tubular boiler using Anzin coal at 17*s.* 9*d.* per ton. The trial lasted 468 hours on the ordinary service of the factory, and the water evaporated per pound of coal was—

In the Belleville boiler............................	8.90 lbs.
In the tubular boiler........................	7.19 „

At the Nice Exhibition, comparative trials of Cardiff coal, Port-de-Bouc briquettes, and Portes briquettes, were, in May, 1884, made in a boiler of

100 nominal horse-power. The coal cost 37*s.* 10½*d.* per ton, and the briquettes respectively 29*s.* and 34*s.* 8*d.* per ton. The grate-area was 39.7 square feet, and the heating surface was 1263 square feet, or 31.8 times the area of fire-grate. Steam was generated under pressures of about 11 atmospheres, or 150 lbs. per square inch, and was blown off direct from the boiler. The leading results are abstracted in the following table:—

Table No. 161.—Belleville Boiler: Trials at Nice Exhibition, 1884.

No. of trial	1	2	3	4
Barometric pressure	30.3 ins.	30.3 ins.	30.3 ins.	29.7 ins.
Exterior temperature	68° F.	68°.9 F.	68° F.	70° F.
Temperature in the boiler-room	72°.5 F.	74°.3 F.	73°.4 F.	76°.1 F.
Duration of trial	3 h. 43 m.	3 h. 29 m.	4 h. 5 m.	4 h. 7 m.
Designation of fuel	briquettes, Bouc	Cardiff coal	briquettes, Bouc	briquettes, Portes
Fuel consumed per hour	498 lbs.	461 lbs.	541 lbs.	482 lbs.
Do. do. per square foot of grate	12.5 „	11.6 „	13.6 „	12.1 „
Ash and clinker, per cent	—	10.7%	10.2%	11.1%
Average working pressure, per square inch	150 lbs.	150 lbs.	150 lbs.	151 lbs.
Temperature of feed-water	68°.5 F.	65° F.	63°.5 F.	75° F.
Water evaporated per hour	4508 lbs. 72.2 cu. ft.	4605 lbs. 73.8 cu. ft.	5009 lbs. 80.3 cu. ft.	4763 lbs. 76.3 cu. ft.
Do. do. per square foot of grate	1.82 „	1.86 „	2.02 „	1.92 „
Do. per pound of coal	9.06 lbs.	9.98 lbs.	9.26 lbs.	9.88 lbs.
Do. do. from 32° F., and at the working pressure of 60 lbs.	8.91 „	9.84 „	9.16 „	9.66 „

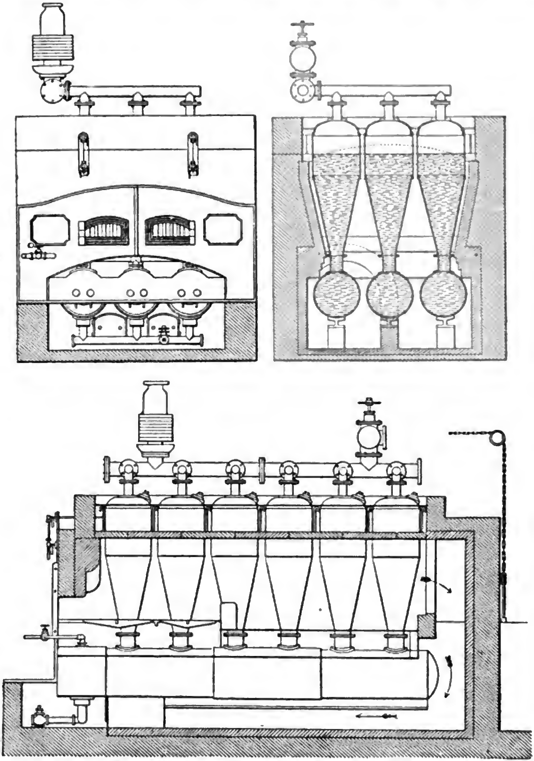
Samples of the feed-water were taken before entering the purifier, and after leaving it. The second samples were allowed to settle for 12 hours, when the solid precipitate was analysed and measured. At 68° F. the following were the respective proportions of the solid matters per gallon of water:—

	Weight of Salts per Gallon.		
	Before Purification.	After Purification.	Proportion Precipitated. per cent.
Carbonate of lime	4.21 grains.	.70 grains.	83
Sulphate of lime	23.56 „	.98 „	96
Magnesia	8.42 „	.84 „	10
	36.19 „	2.52 „	93

THE SHEPHERD SECTIONAL STEAM BOILER.

The Shepherd sectional boiler, constructed by Messrs. James Watt & Co., consists of two or three rows of independent upright vessels, partly conical and partly cylindrical, fastened respectively on two or three horizontal cylinders or tubes. The design of a triple-row boiler is shown in figs. 351. The bottom cylinders, three in number, are each 2 feet in diameter, and 16 feet in length, supported on cast-iron seats laid on brickwork, at 2 feet

4 inches apart between centres. On each cylinder six conical vessels are fastened by 6-inch necks. They widen upwards from 6 inches to 2 feet in diameter at the cylindrical head; and they are 6 feet high, or, including the necking, 6½ feet high. They are placed at 2 feet 4 inches apart between centres. A 3-inch necking is fixed on the top of each conical vessel for the escape of steam, which is collected by 3-inch transverse pipes, one to every transverse row of three vessels, from which it is delivered into



Figs. 351.—The Shepherd Sectional Steam Boiler. Scale 1/64th.

one 6-inch main pipe, on which the steam stop-valve and the safety-valve are fastened.

A 3-inch hand-hole is cut in the top of each conical vessel, closed by a cover and a single bolt and nut. The fire-grate is placed between and surrounding the conical vessels at the lower part, with a brick bridge at the back; and the burnt gases pass between and round the conical surfaces to the back of the furnace-chamber. They there descend, and return to the front between and under the bottom cylinders, at one side, whence they pass again to the back and into the chimney.

To guide the draught, a flooring or covering of iron plates is laid, forming the bottom of the flueway to the back, and shielding the connections with the bottom cylinders from the action of the gases. For the same purpose another layer of iron plates is placed at a lower level, forming the tops of the bottom flues. The roof of the flueway or furnace-chamber consists of fire-clay blocks laid in just under the water-level in the conical vessels. Above this roofing a covering of iron plates is laid in, round the heads of these vessels.

The working pressure is 150 lbs. per square inch; the hydraulic test-pressure is 300 lbs. per square inch.

The length of the boiler over all, exclusive of brickwork, is 16 feet, the width is 6½ feet, and the height above the floor-line is 9 feet. The area of fire-grate is 26.18 square feet; and the heating surface is 529 square feet, or 20.20 square feet per foot of grate-area. The capacity for water is 293 cubic feet; and for steam, 87 cubic feet; together 380 cubic feet. The weight of the boiler complete is 13.17 tons.

Many Shepherd boilers have been sent to India and other countries where there is great difficulty of transport.

Two days' trials of this boiler were conducted by Mr. Henry Hiller in March, 1879. They lasted nine consecutive hours on each day. The coal used was Burgey, of good quality, fairly round, with not much slack.

Table No. 162.—SHEPHERD SECTIONAL STEAM BOILER: RESULTS OF TRIALS.

No. of trial	1	2	Mean
Average working pressure in the boiler, per square inch	79 lbs.	62½ lbs.	71.25 lbs.
Coal consumed per hour	339 „	364 „	352 „
Do. do. per square foot of grate-area	12.95 „	13.90 „	13.43 „
Ash and clinker, per cent	16%	11.4%	13.7%
Temperature of feed-water	50°.7 F.	50°.7 F.	50°.7 F.
Water evaporated per hour	2827 lbs. 45.3 cu. ft.	3200 lbs. 51.3 cu. ft.	3014 lbs. 48.3 cu. ft.
Do. do. per square foot of grate-area	1.73 „	1.96 „	1.85 „
Do. do. per pound of coal	8.33 lbs.	8.79 lbs.	8.56 lbs.
Do. do. from and at 212° F.	9.98 „	10.48 „	10.28 „
Temperature of gases in the flue to the chimney	—	590° to 700°	620°

The test commenced each day with a thin clear fire, and was ended with the fire in the same condition. The average depth of fuel on the first day was about 6 inches, on the second about 8 inches; each side of the furnace being fired alternately. The fires were cleaned once in the middle of each day, and at the end of the first day's test. On the first day the opening of the damper was regulated to meet the varying requirements for steam; on the second day the damper was kept in one position for the whole time.

MILLER'S CAST-IRON SECTIONAL STEAM BOILER.

The cast-iron steam boiler, figs. 352 to 354, designed by Mr. J. A. Miller of Boston, U.S.A., was introduced into England by Mr. John Laybourne in 1869.[1] It is constructed of a series of separate cast-iron sections, joined together at the base of each by flange-joints. The sections are of two patterns only, each of comparatively small size, so as to contain only a small quantity of water. Those at the front end are ∩-shaped tubes, which, side by side, form a succession of arches over the fire-grate, as shown in figs. 353 and 354. The rear sections, fig. 352, consist each of five vertical tubes cast in one piece, united by a transverse horizontal tube at the bottom and one at the top; and finished at the top with a flange-joint, upon which a cover is fixed. The sections are bolted together at the bottom by flange-joints, forming continuous longitudinal tubes at the bottom of the boiler. The tubes of the fire-box sections are 7 inches in diameter inside, with 2 feet 4 inches width of arched opening, which defines the width of the fire-grate, as shown in fig. 353.

The vertical tubes of the rear sections are taper in form, 4 inches in diameter inside at the bottom, and 6 inches at the top, and 2½ feet long; averaging 2 inches clear. The tubes of consecutive sections are flanged zigzag with each other so as to break up and mix the hot gases. The castings are ½ inch thick; the rear sections weigh 10 cwts. each, and the front arched sections, 5 cwts. each.

Circulation of water is maintained through the tubes by means of mid-feathers and internal tubes, as shown. Steam is collected in a horizontal 10-inch main above.

The experience of 2½ years of continuous working showed that when the boiler was periodically blown out under pressure, sediment did not injuriously collect in any part exposed to the heat of the fire or the flue, whilst the cast iron bore the heat of the fire without injury.

An automatic feed apparatus, designed by Mr. Berryman, is shown in fig. 352. A hollow cast-iron ball, suspended from a counterpoised lever, is connected by two ½-inch pipes, at the top and the bottom of the ball, with the boiler at the high-water level and a lower level. When the level falls below the end of the upper pipe, steam flows into the ball, which is at its highest level, and the feed-cock is opened to the boiler. When the water level rises and closes the upper pipe for steam, the steam in the ball condenses, the ball fills with water and falls, shutting off the feed; and so on.

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[1] See "Description of Miller's Cast-iron Steam Boiler," by Mr. John Laybourne, in the *Proceedings of the Institution of Mechanical Engineers*, 1871; page 263.

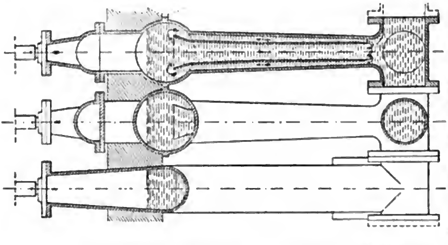


Fig. 354.—Front and Rear Sections. Scale 1/20th.

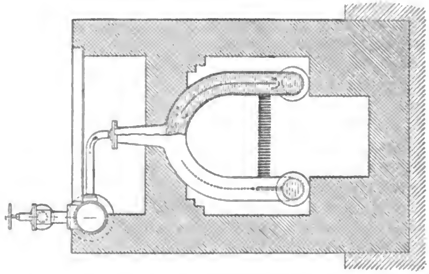


Fig. 353.—Front Sections. Scale 1/40th.

Miller's Cast-iron Sectional Steam Boiler.

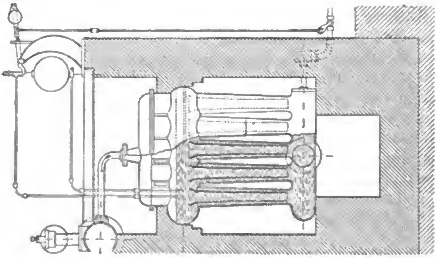


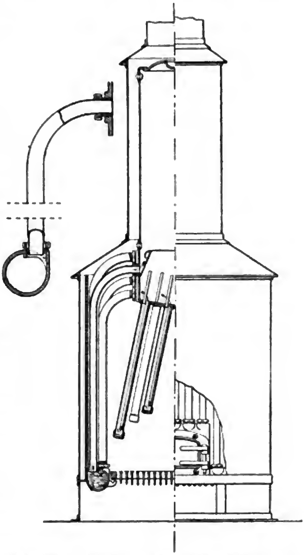
Fig. 352.—Rear Sections. Scale 1/40th.

The effective heating surface of each rear section of the boiler is 23 square feet, considered equivalent to 2 horse-power; and that of each arched section, above the level of the fire-grate, is 7 square feet, also of 2 horse-power.

A boiler of this class, erected at Isca Foundry, Newport, had eight arched sections, and 14 rear sections, giving 378 square feet of heating surface, with 17 square feet of fire-grate surface, bearing the ratio 22.2 to 1. In a series of experiments made upon this boiler, by Mr. Joseph Tomlinson, with several kinds of Cardiff steam coal, the rate of evaporation ranged from 37 cubic feet to 43.3 cubic feet of water per hour; and the efficiency ranged from 9.37 pounds to 10.15 pounds of water per pound of coal; the coal consumed per square foot of fire-grate being from 13.53 pounds to 16.77 pounds per hour. The temperature of the feed-water was from 56° to 70° F. The temperature in the chimney-flue varied from 500° to 575° F. The fire-bars were ½ inch apart. This boiler has been removed, in consequence of the priming of the boiler when worked with bad water.

THE WARD SECTIONAL STEAM BOILER.

The Ward boiler is the invention of Mr. Chas. Ward, Charleston, U.S. As shown in figs. 355, it is a vertical boiler, cylindrical in outline. The foundation of the boiler is a circular hollow tube of cast steel, 3¾ inches in diameter internally, ⅜ inch thick, formed to a circle 35 inches in diameter, extreme. This ring is supported by the ash-pit, and it carries the fire-grate, which is 26 inches in diameter. The furnace is circular, and is formed by vertical tubes of wrought iron, placed circularly, zigzag in plan, their axes being alternately 16¼ inches and 14¼ inches from the axis of the boiler. They are screwed by their lower ends into the base-ring, and into a drum at the upper part. They are 72 in number, 1.38 inches bore, .14 inch thick, and from 3 feet 11¼ inches to 3 feet 2 inches in length. The drum forming the upper part of the boiler is in two pieces, the lower piece being of cast steel, ⅜ inch thick, 14¾ inches in diameter externally; the upper part of wrought iron, 14 inches in diameter outside, riveted to the lower. The total height of the drum is 3 feet 10½ inches.



Figs. 355.—Ward Sectional Steam Boiler. Scale 1/24th. With Detail of Tube.

Into the bottom of the drum three circular rows of 1.38-inch hanging tubes are screwed, closed at the lower ends by screwed caps, numbering 16, 8, and 4 tubes successively; and 26½ inches, 23 inches, and 21 inches in length. They have each a 9/16-inch brass tube, fixed within them for promoting circulation.

Table No. 163.—WARD SECTIONAL STEAM BOILER: RESULTS OF TRIALS, 1884.

FUEL............	Semi-bituminous.	Semi-bituminous, with Blast in Chimney.	Anthracite.	Anthracite, with Blast in Chimney.
Working pressure of steam per square inch	Atmospheric.	160.7 lbs.	Atmospheric.	16.82 lbs. to 240.96 lbs.
Fuel consumed per hour	49.7 lbs.	230.5 lbs.	29.1 lbs.	60.4 lbs.
Do. do. per sq. foot of grate..........	13.5 lbs.	62.5 lbs.	7.9 lbs.	16.4 lbs.
Temperature of feed-water	66°.1 F.	69°.6	65°.9	67°.3
Water evaporated per hour	380.0 lbs.	1219.6 lbs.	217.2 lbs.	408.1 lbs. to 360.9 lbs.
Do. per pound of coal as from and at 212° F..................	8.81 lbs.	6.34 lbs.	8.59 lbs.	8.28 lbs. to 7.26 lbs.
Heat utilized in evaporation, in parts of the total heating power of the fuel	64.4%	46.0%	71.7%	66.7% to 60.2%

The boiler is inclosed by two concentric sheet-iron casings, ¾ inch apart, the inner casing being lined with 3/16-inch asbestos board as a non-conductor; 38 inches in diameter outside. The chimney is of sheet iron, 11¼ inches in diameter inside, 24 feet high above the level of the fire-grate.

The total external heating surface is 145.8 square feet; calculated internally it is 121.6 square feet. The area of fire-grate is 3.69 square feet. The weight of metal in the boiler, excluding the chimney, is 1652 pounds; of the asbestos, 30 pounds; of water at 60° F., 248 pounds; together, 1930 pounds.

This boiler was tested in the New York Navy-yard,[1] in June, 1884, with the results shown in table No. 163, above.

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[1] See *Report made to the Bureau of Steam Engineering*, U.S.; September 10, 1884.

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